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CONTENTS MARCH 1951

THE RELATIVE MERITS OF CENTRIFUGAL AND AXIAL COMPRESSORS FOR AIRCRAFT GAS TURBINES
E. S. MOULT, B.Sc., Ph.D., F.R.Ae.S., M.I.Mech.E., and H. PEARSON, B.A., A.F.R.Ae.S.

METAL ADHESIVE PROCESSES *F. H. PARKER*

METHOD FOR DESIGNING WIND TUNNEL CONTRACTIONS
R. HARROP

A NOTE ON THE BUCKLING OF STRUTS
HAROLD LURIE, Ph.D.

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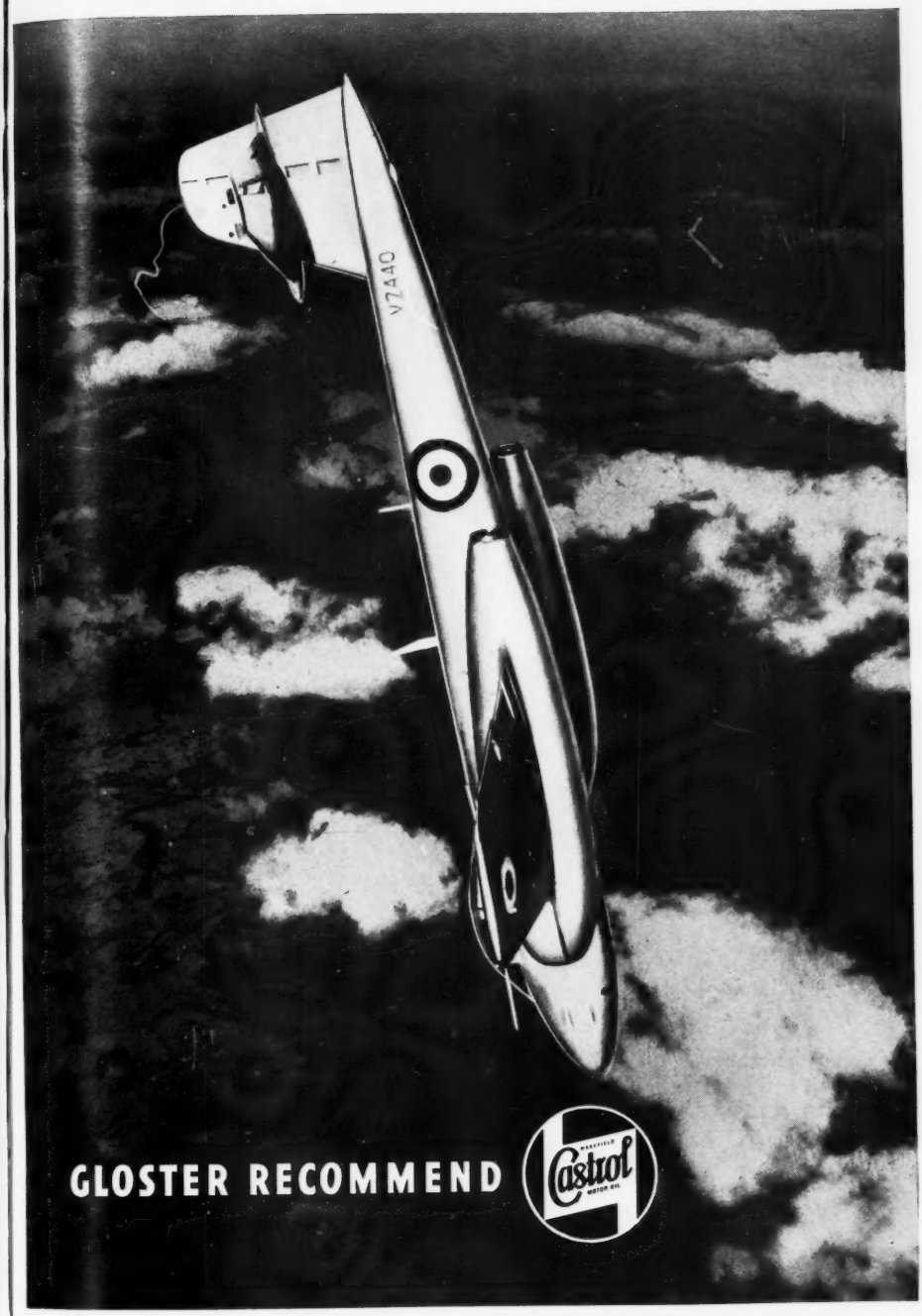
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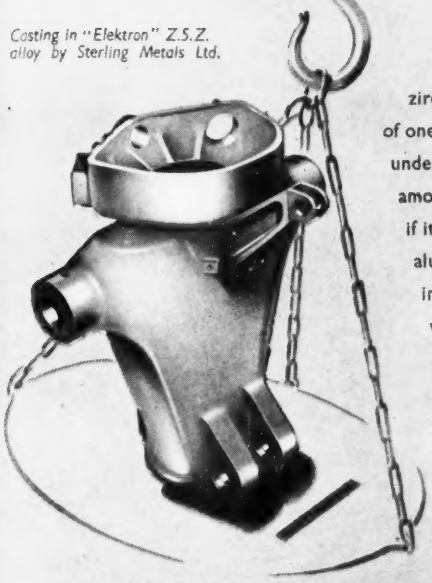

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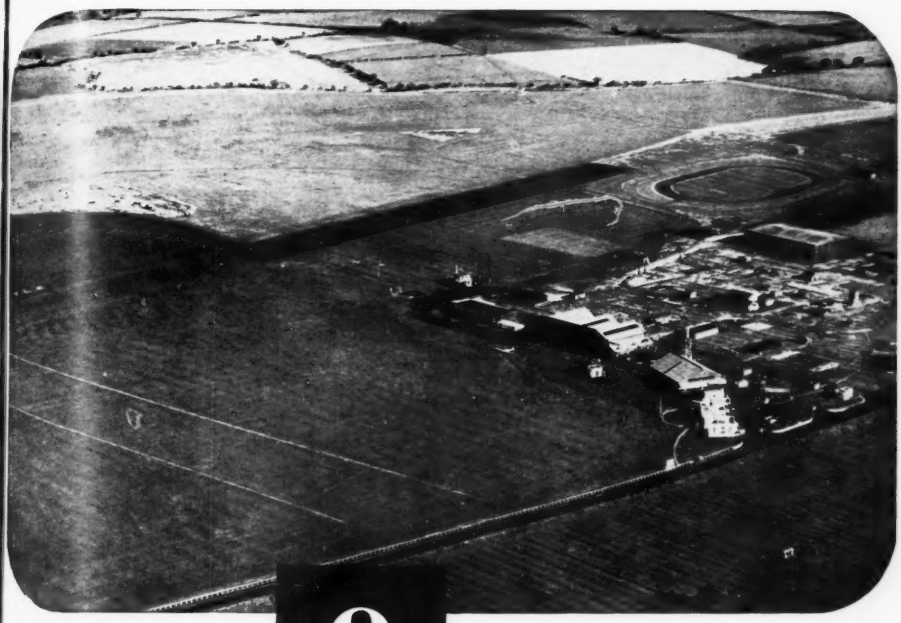

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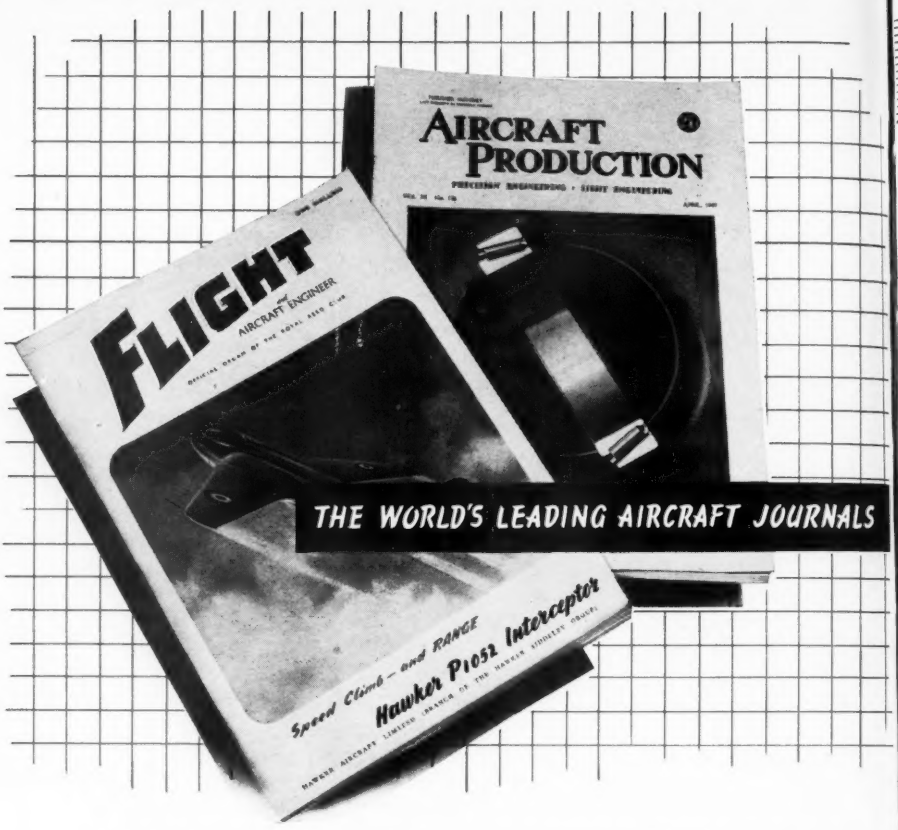

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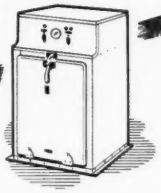

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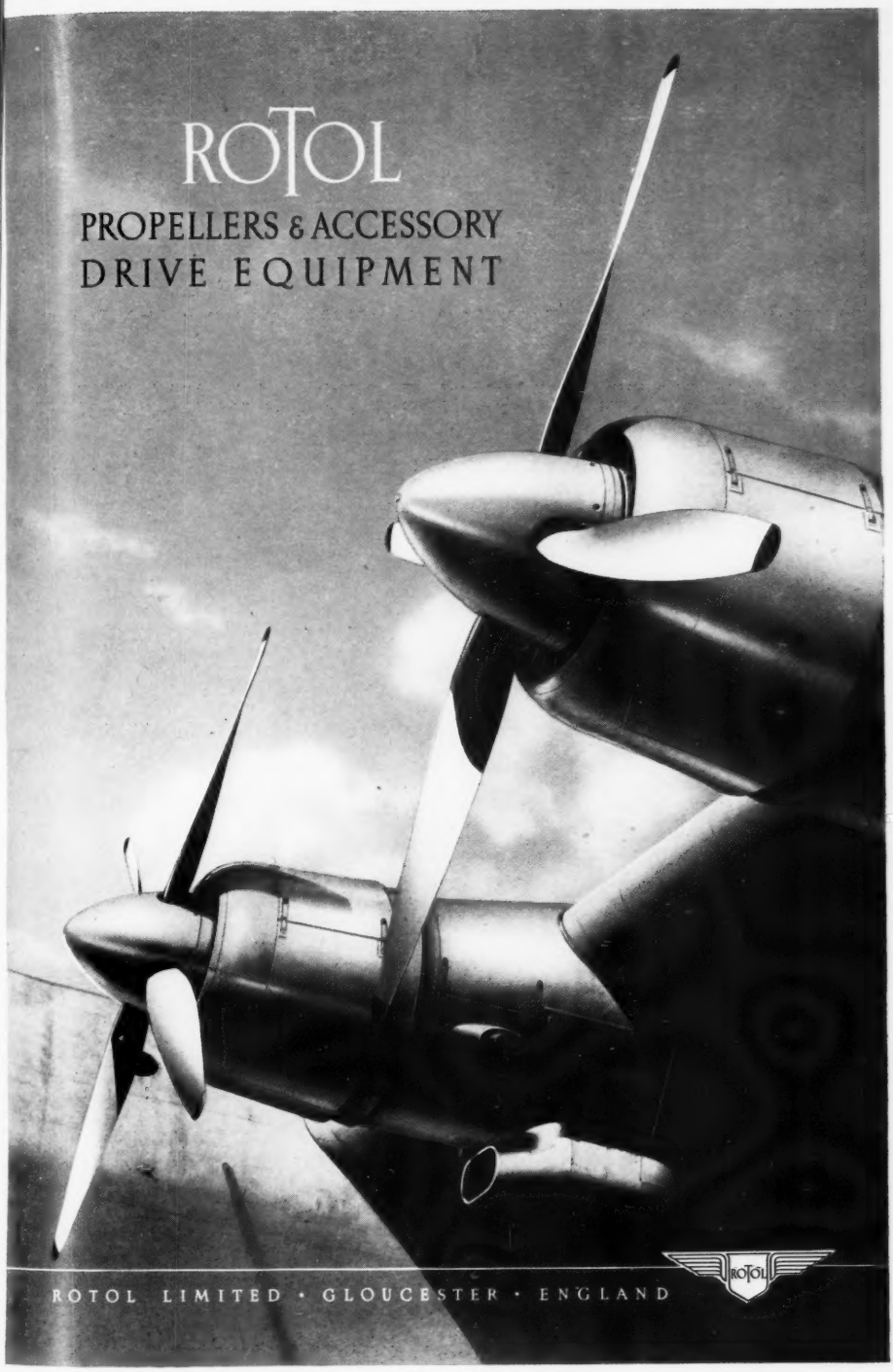
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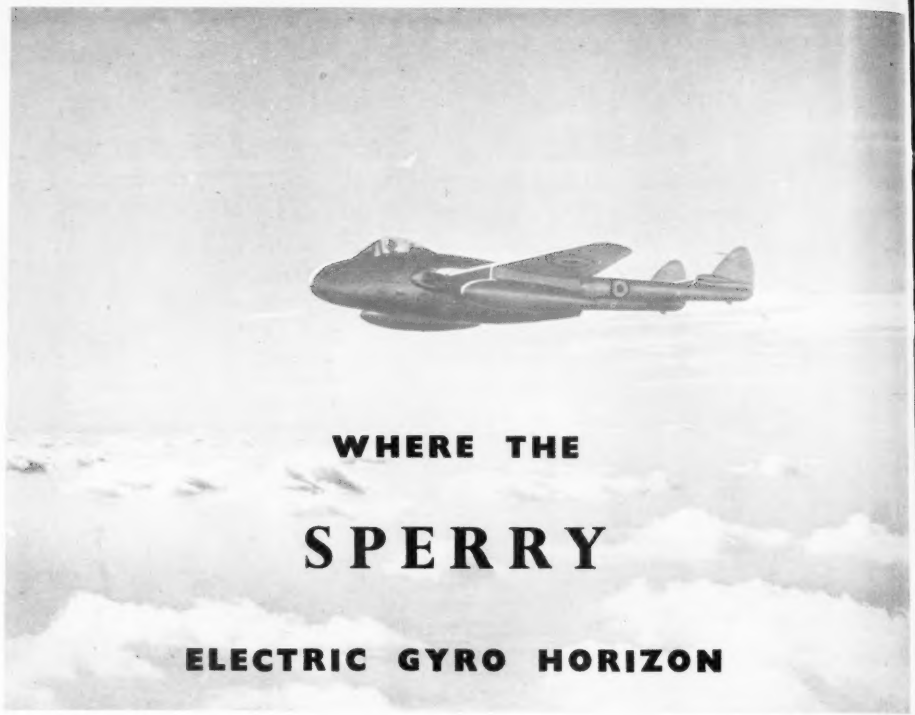

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THE RELATIVE MERITS OF CENTRIFUGAL AND AXIAL COMPRESSORS FOR AIRCRAFT GAS TURBINES

A DISCUSSION LED BY

E. S. MOULT, B.Sc., Ph.D., F.R.Ae.S., M.I.Mech.E.

and

H. PEARSON, B.A., A.F.R.Ae.S.

A Discussion, at which four short papers were presented, was held by the Royal Aeronautical Society on 7th December 1950 at the Institution of Civil Engineers, Great George Street, London, S.W.1. Major G. P. Bulman, C.B.E., F.R.Ae.S., President of the Society, introduced the two main speakers, Dr. E. S. Moult, B.Sc., F.R.Ae.S., M.I.Mech.E., Chief Engineer, The de Havilland Engine Co. Ltd., and Mr. H. Pearson, B.A., A.F.R.Ae.S., Chief Performance Engineer, Gas Turbine Department, Rolls-Royce Ltd.

Dr. Moult was supported by Mr. J. L. Brodie, A.F.R.Ae.S., M.I.Mech.E., Hon.M.L.A.E., Director in charge of Engineering Division, de Havilland Engine Co. Ltd., and Mr. Pearson by Mr. A. C. Lovesey, O.B.E., B.Sc., F.R.Ae.S., Chief Development Engineer, Rolls-Royce Ltd.

THE CENTRIFUGAL COMPRESSOR — Dr. MOULT

The suitability of a piece of mechanism for a given job depended both on the nature of the job and on the solution offered. In aviation they had to consider performance, time of development, cost in production and serviceability in its widest sense. It was proposed to review these different aspects as they applied to the compressors for aircraft gas turbines.

The function of the compressor was to take in the appropriate amount of air required by the engine and to compress it through the chosen pressure ratio as efficiently as possible. For this purpose the centrifugal compressor was unbeaten in the sense that it gave the highest pressure rise in a single stage. On this fact depended its simplicity and robustness.

As regards efficiency, it was agreed at the outset that the axial compressor led by a few points at its design condition. Nevertheless, its pressure and volume characteristics were sharply tuned and it was more difficult to hit and to hold the optimum. Axial blading was very sensitive to changes of incidence and in flight a wide range of loads and speeds must be covered. In the interests of stability there must be some sacrifice of efficiency at certain points.

By contrast, the centrifugal machine had more accommodating characteristics and experience indicated that it would maintain a high efficiency over its whole working range without the addition of any special stabilising devices. Again there was strong evidence that the centrifugal compressor was less susceptible to the effects of Reynolds' number than the axial, which might lose five or 10 per cent. in efficiency between sea level and 40,000 ft. altitude. The axial could not

help this, but the point he wanted to make was that there was a world of difference between what might be called the effective operating performance and a figure of thrust or consumption taken on a test bed.

Figure 1 compared the pressure/volume characteristics of two actual compressors of similar pressure ratio. The mass flows had been adjusted to correspond at 100 per cent. r.p.m. On the same curve was indicated a typical engine operating curve, firstly with the aircraft at rest (ram ratio=1) and secondly at a forward speed giving a ram ratio of 1·5.

It would be noted that the axial characteristics were steeper than those of the centrifugal so that there was a greater change of pressure and of efficiency as the working point moved in accordance with the conditions of operation. Moreover, it was not permissible to operate a compressor to the left of the surge lines and the margin between this limit and the operating point was less for the axial than for the centrifugal. Conversely there must be a greater drop in the performance of an axial for an equal margin of stability. This margin of stability was of great importance for high altitude operation and for conditions of rapid acceleration.

In the matter of size, the diameter of the conventional centrifugal compressor for a given rate of air flow was usually larger than that of the corresponding axial unit. This might or might not matter according to the arrangement of the installation. For example, it was not possible to reduce the diameter of the fuselage of the Vampire which was already about the minimum to give the necessary room for pilot, guns and so on. Again, in the case of the Comet the drag of the four nacelles was less than two per cent. of the total drag of the whole aircraft and the room for improvement in this direction was obviously negligible. Recent developments showed that it was possible at least to halve the frontal area of existing centrifugal compressors, while retaining the same through-put and so the same engine output in terms of thrust or b.h.p. In fact, the present position was that the designer had reached the stage where he was embarrassed not by the size of his

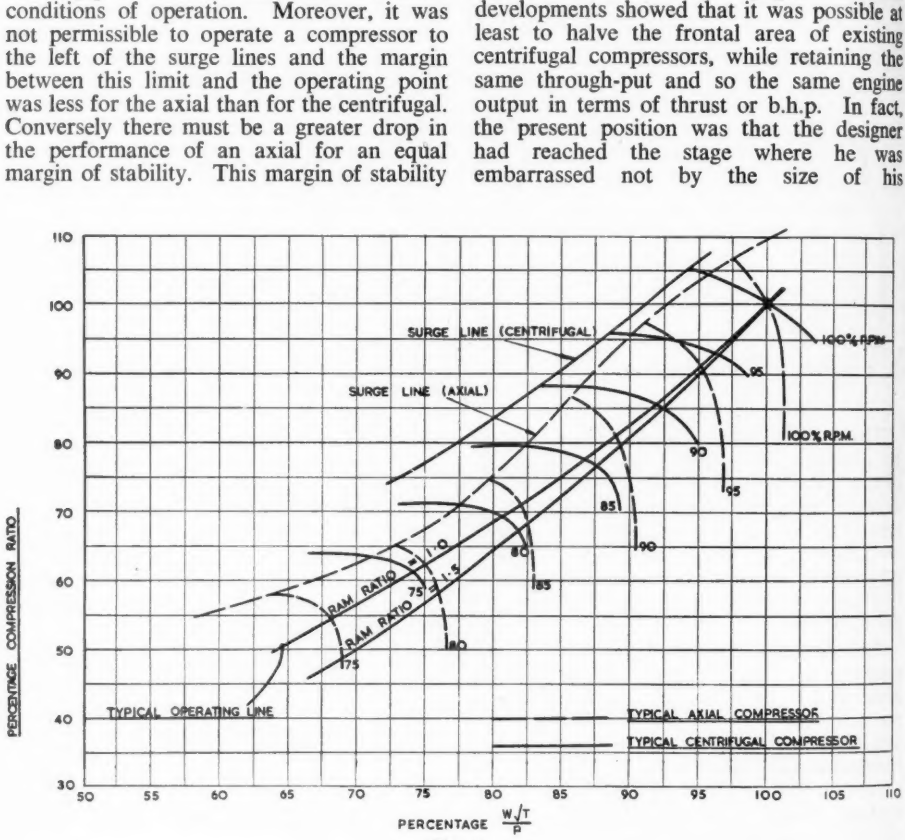


Fig. 1.

Comparison of compressor characteristics, typical axial and centrifugal compressors of equal pressure ratio.

compressor, but with the problem of housing a turbine and combustion system capable of dealing with the available mass flow.

Figure 2 compared in outline a typical axial engine and a possible centrifugal engine of equivalent air mass flow and thrust output. The centrifugal engine indicated was based on the development work mentioned and it would be noted that the difference in bulk was negligible. The diagram represented what was currently possible and in neither case indicated the ultimate that might be achieved.

It had been said that the design of a centrifugal compressor was not an exact science. Be that as it may, there existed a wealth of experience with centrifugal compressors on which the designer might draw with confidence and it was seldom that any major change in design was necessary during development. The aerodynamics of an axial-flow compressor might be fairly well established for the early stages of compression, but there was a pronounced deterioration of flow in the later stages. Moreover, the best results from a gas turbine were attained only when the compressor and turbine were so "matched" that both components attained their peak efficiency simultaneously. The sharp characteristics of the axial made this matching process more difficult and it was not uncommon for an axial compressor to be re-bladed many times before it attained its designed performance and was free from other vices.

All compressors were subject to an unstable condition of operation known as "surging" if their mass flows were reduced below a certain critical value. It was perhaps unfortunate that the point of maximum efficiency lay near the surge line for most compressors and, with the axial's narrow operating band, it was difficult to steer a safe course between instability on the one hand and good efficiency on the other.

To pressurise the cabin or to de-ice the airframe it was convenient to bleed off air from the compressor in quantities that varied with the circumstances. This was straightforward up to about 10 per cent. flow with the centrifugal, but was much more critical with an axial.

Anything that tended to vary the mass flow from the chosen value led to trouble and so they had blow-off valves, variable incidence guide vanes, variable propelling nozzles and other strange devices that were used as palliatives, or even as prime essentials. He had seen an axial suddenly stall and stop because a little ice formed over the intake guard and partly obstructed the air flow. He was told that another axial emitted a ball of flame and stopped if it were opened up suddenly during an approach.

Those were serious problems which might not be insuperable—but they did add to the development time, which was always a vital question in aviation. Those were a few reasons why in Great Britain there were in service to-day thousands of centrifugally-engined aircraft compared, at most, with a few dozen aircraft with axial-flow compressors.

In matters of manufacture and serviceability there were others more qualified to speak than he. On the basis of similar outputs the cost of the axial flow engine, whether reckoned in terms of cash or of man hours, was perhaps twice the cost of the corresponding centrifugal engine.

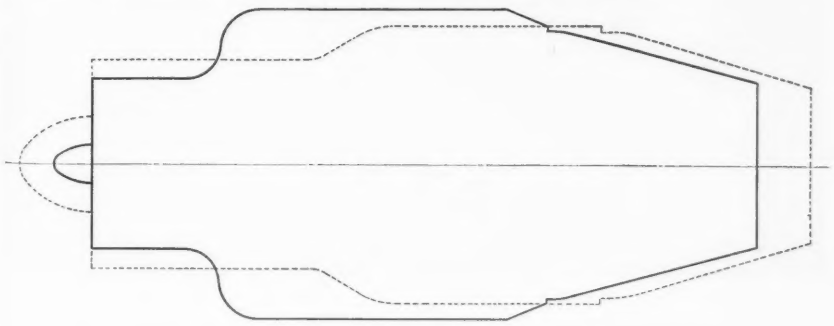


Fig. 2.
Comparative outline drawings of axial and centrifugal engines of equal thrust.

Undoubtedly this price difference lay mainly in the problems of manufacture and the satisfactory retention of large numbers of fixed and moving compressor blades. Extremely fine limits of accuracy were essential for good performance.

By the term serviceability was meant the ability to continue the allotted function without loss of efficiency or danger of mechanical breakdown. Whatever they might wish, aircraft power plants could not be wrapped in cotton wool for their whole working lives; they must be able to withstand the variations of atmosphere and climatic conditions, in addition to the thermal, mechanical and aerodynamic loads to which they were submitted.

It was here that the centrifugal compressor scored so heavily by virtue of its simple and robust construction. The sturdy impeller was practically immune to the effects of vibration, dirt, snow or ice and it continued to give unimpaired performance after many hundreds of hours' service. For example, a Goblin engine was run on a series of combat

Fig. 3.

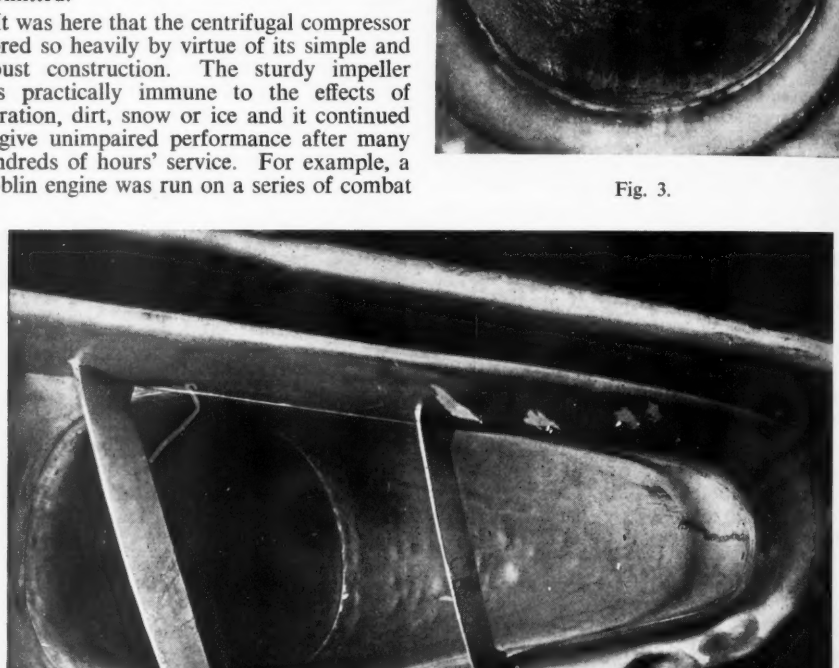


Fig. 4.

This photograph illustrates the damage suffered by the 2 in. hailstones hitting a Vampire in flight in Switzerland. The pilot reported no irregularity of operation and the impeller, which was subsequently examined, suffered no damage at all and is still in use.

cycles for over 1,200 hours and at the end gave performance figures showing negligible fall-off and well within the acceptance limits for a new engine. An axial compressor with its precise aerofoil surfaces was much more susceptible to the effects of dirt and abrasion. Losses of five and even 10 per cent. had been experienced in the course of a type test of 150 hours' duration.

No one would deny that the vibration problems in the blading of an axial compressor were much more complex and difficult of solution than those in a centrifugal impeller. Each blade had many forms of vibration and there were many types of blade even in a single compressor. To dodge all resonant frequencies on all blades in the whole speed range was well-nigh impossible. In fact, the problem of holding those frequencies and of exploring all possibilities during the period of development testing itself was a formidable one. Moreover, there was the problem of root fixation and its stress concentrating effects on both blades and discs.

Experience had shown the centrifugal compressor to be particularly resistant to the effects of snow and ice. Because of its speed and working temperature ice would not stick to the impeller in quantities likely to affect performance: in fact it would digest even sizeable lumps of ice and would only object if the inhaled masses were of such size and weight as to cause serious bodily harm.

Figure 3 was a photograph taken of the interior of a long aircraft intake fitted to a Ghost engine in the course of ice protection tests. Heavy accumulations of ice would be noted on the lip of the intake and as a thick longitudinal patch. The piece of this patch which could be seen broken away was about 2 in. thick and 2 ft. long. Presumably it achieved a considerable velocity at the moment of impact with the rotating impeller.

A lump of ice of six or seven pounds weight had been inhaled some time ago by a Ghost engine. Apart from causing a momentary hiccup there was no permanent damage and the particular impeller continued in regular use. They had experience also of damage suffered by a Vampire when flying into a storm over Switzerland when hailstones of over two inches in diameter seriously battered the airframe and intake ducts, but the impeller was undamaged and the engine continued to function normally (Fig. 4).

In Canada a squadron of Vampires in a formation take-off flew through a dense flock of hundreds of birds and continued on its mission. The official report stated that the engines operated normally after a short period of rough running, and concluded with the words: "It appears therefore that the Goblin II engine can swallow birds without damage or effect on performance." He hesitated to think what could have happened to present-day axial compressors under similar circumstances.

THE AXIAL COMPRESSOR — Mr. PEARSON

INTRODUCTION

There were probably not a few among the audience who had had close connections with the development of the axial compressor over a much longer period than he, and who had consistently advocated it when much more work was being done on centrifugal development. To them he could only apologise and quote that "there is more joy in heaven over one sinner that repenteth than over 99 righteous persons."

He would like to congratulate Dr. Moult for his masterly rendering of what was obviously a bad case. He hoped the audience would not allow their judgment to be swayed from the main facts of the case by the pure debating skill of either side.

In certain respects he was specially fitted to pronounce a balanced judgment in this discussion because he had been connected with the development of both types of compressor, and indeed his firm was still working on both types. On the other hand his opponent's firm seemed so far to have made up their minds on one line only, but it would not surprise him as a result of this discussion, to find them coming out with an axial compressor in the near future.

He had been a great believer in the past in the centrifugal compressor, and he considered that belief had been well justified in the remarkable achievements that had been made in the jet engine from this development. He had no doubt that at the present time there

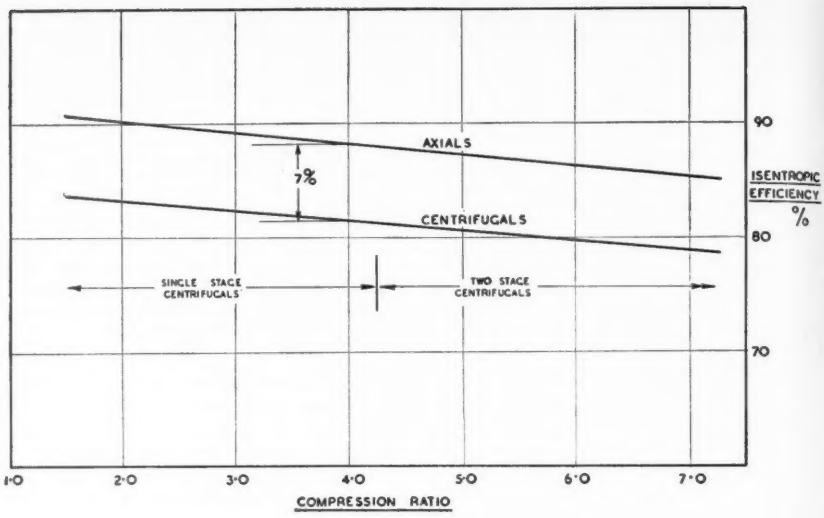


Fig. 1.
Variation of maximum efficiency with compression ratio.

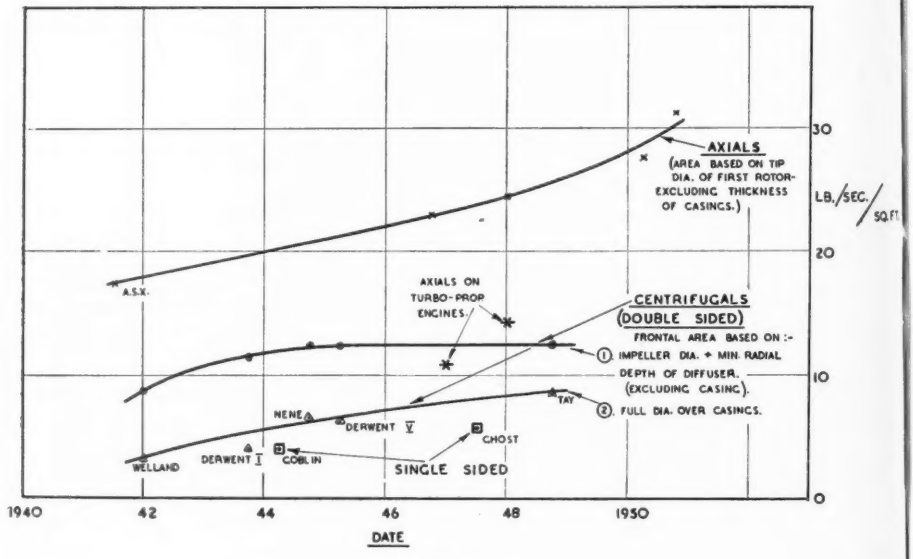


Fig. 2.
Mass flow per unit frontal area.

were many uses for both types of turbine engines with axial or centrifugal compressors, but in reviewing the possibilities of the greatest possible performances and especially looking to the future development of those compressors, it had been increasingly borne in on him in the past few years that the axial compressor easily had the ultimate advantage.

In considering the characteristics of the compressor for aircraft gas turbine work, it seemed to him that the following factors had to be considered:—

(*a*) Efficiency
(*b*) Air Mass Flow per Unit Frontal Area
(*c*) Weight and compactness for given duty
(*d*) Flexibility
(*e*) Reliability
(*f*) Cost.

He hoped in the limited time available, to deal with the first four and that in supporting him Mr. Lovesey would deal, among other matters, with the last two factors.

(*a*) *Efficiency*

It was generally a by-word that axial compressors were the more efficient, but to emphasise this point he had prepared Fig. 1 to show the summary of their data for both types at various compression ratios. The mean curves were based on a wealth of test

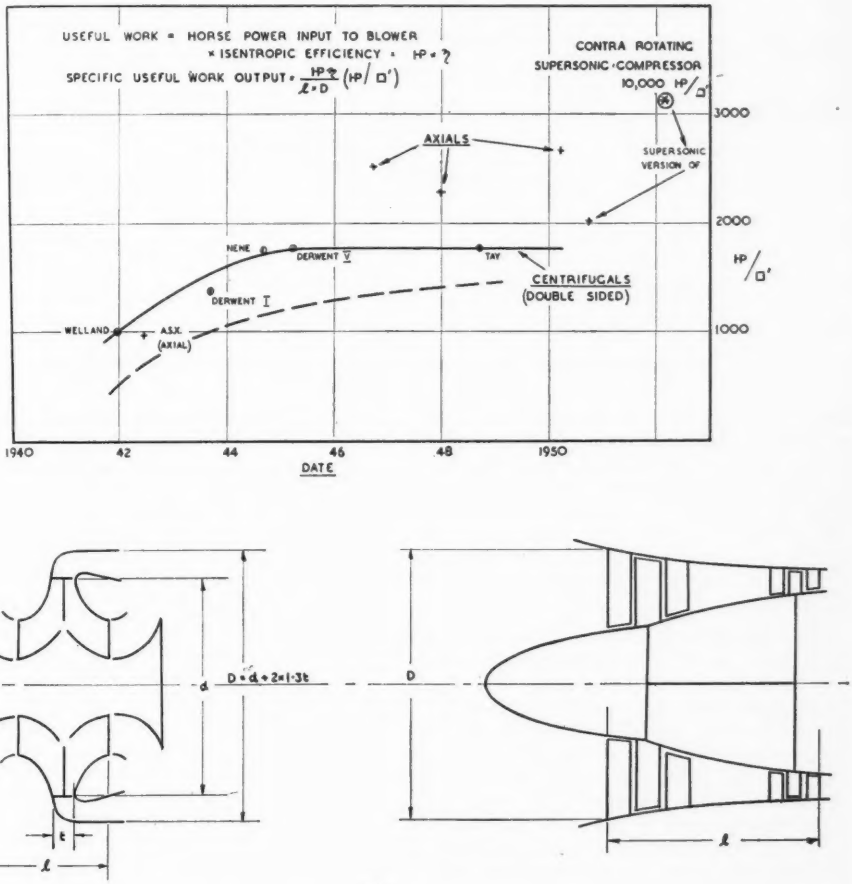


Fig. 3.
Specific "useful work" output.

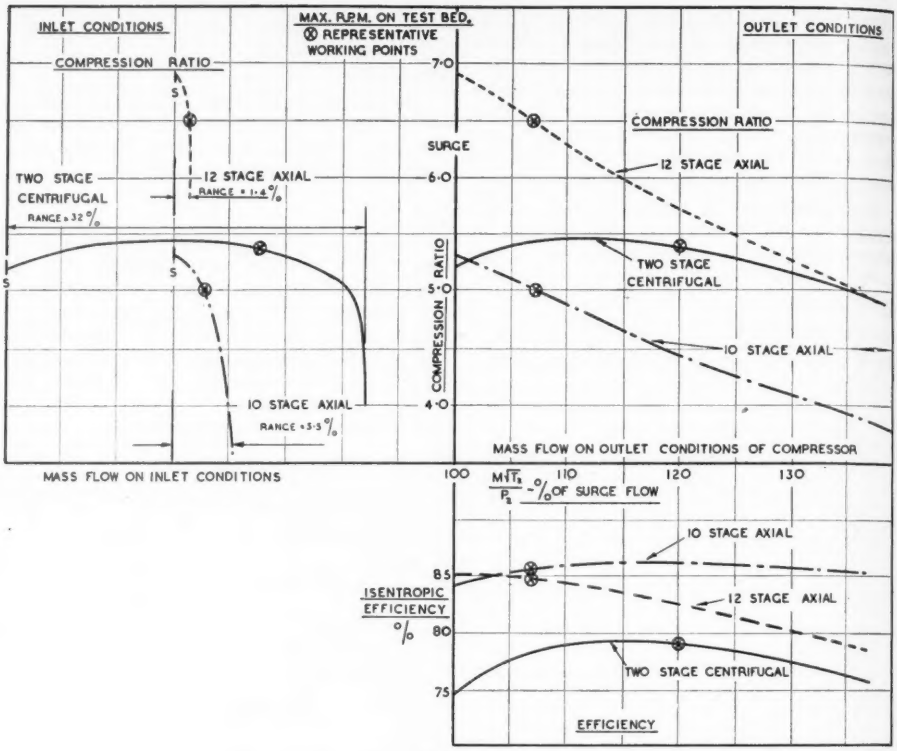


Fig. 4.
Comparative range of axial and centrifugal compressor characteristics.

data of their own and on published results of other firms, but in this figure and in his others security and simplicity had forced him to remove many otherwise interesting points from the curves. It was their general conclusion that at almost any compression ratio up to at least 7 : 1 they could reckon on 7 per cent. better efficiency from the axial compressor.

Why should this inherently be so? The centrifugal people had always been saying how well they could do if only they could concentrate more work on basic problems of the blower, *e.g.* the inducer vanes and the diffuser losses and so on. Far more work in the past ten years had been done on the centrifugal than on the axial and their experience pointed to the fact that this gap would remain, assuming a fair proportion of work on both sides. The reason appeared to him to be inherently in the method of doing the work; in the axial the air was given a small amount of velocity and this was then diffused to pressure and this process repeated many times. In the centrifugal the work was all done at one go, so to speak. This would be all right if all the losses remained proportional to the velocity squared, but unfortunately the velocities at the impeller tips were so high that the consequent diffuser losses were out of all proportion. To reduce those velocities so that the losses were reasonable would require so much more work from the centrifugal as certainly to make it non-competitive with the axial compressor.

Furthermore, the normal arrangements for the centrifugal compressor were such as to make the losses in the components higher for the same incident velocities as the axial compressor. For example, the deflection required in the inducer vanes was usually about 60° and the deflection in the diffuser some 75°.

To escape from those limitations required changes in some other components, such as

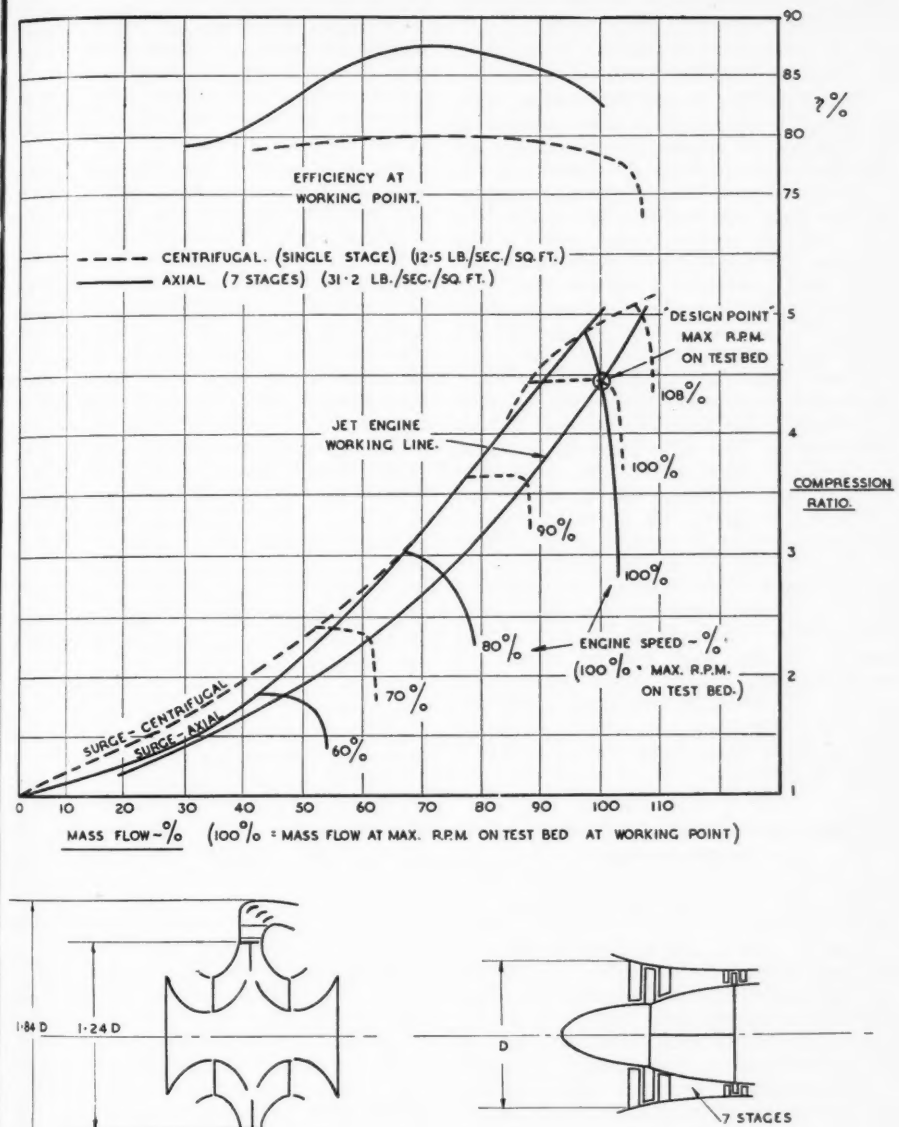


Fig. 5.
Characteristics of centrifugal and axial type compressors designed for the same compression ratio.

the impeller, which would decrease their efficiencies and attempts in this direction had certainly met with little success.

(b) Flow per Unit Frontal Area

The advantages of low frontal area might be arguable when engines were installed in the fuselage of already large aircraft, but with the great increase in speed and the possibility of supersonic flight, and particularly in wing installations, the accent was quite definitely on low frontal area. It was here that the axial had its greatest advantage, which came from the fact that the whole area of the compressor was used for intake flow, whereas this could not be the case for the centrifugal. Fig. 2 showed a comparison between actual axial and centrifugal compressors, in this respect plotted against date. This showed the axial compressor to pass some three or more times as much air as the centrifugal. Here again the centrifugal experts were constantly saying what they could do if only, etc., etc. One great claim was the reduction in diameter if the diffuser were carried out in an axial direction. He had allowed for this therefore by calculating the frontal area, assuming that the air was turned over directly from the tips of the impeller in the least possible space. It would be seen from the extra curve that this at best only made the centrifugal about half as good as the axial. It must be conceded then that the axial compressor won hands down on frontal area.

(c) Weight and compactness for a given duty

The axial compressor might be slimmer; it was usually longer up to the limit of compression ratio of a single stage centrifugal. It had been claimed that this led to a more compact centrifugal and a reduced weight for this type. Weight was a difficult subject to argue about logically, being so subject to the ingenuity of the designer and the metallurgical techniques available and also, he had found it difficult to obtain real information from a variety of sources. The bulk was more a subject of calculation, being largely aerodynamically limited and, in his opinion, it was the ultimate controlling factor in weight. If volume occupied by a compressor for a given power were considered, this would be a useful method of comparison. Unfortunately as many of them knew, the ratio was directly susceptible to the square cube law when scaling and this would not be satisfactory when comparing compressors over a large range of size.

On Fig. 3 he had plotted the useful power divided by the cross sectional area, *i.e.* the area of the compressor seen on a drawing and it would be appreciated that this was a somewhat ingenious method of comparing compressors independently of size.

Figure 3 showed that the two types of compressor were on a much more even footing, but later developments in axials were certainly superior by a factor of 50 per cent. Here again the superiority of the axial compressor was evident. Attention was beginning to turn to the possibilities of the supersonic axial compressor; in this case it was probable that some large sacrifice in efficiency would have to be made (say to make them comparable to the centrifugal in this respect), but without any sacrifice on frontal area, extremely high values of specific useful power might be achieved, as indicated hopefully on the figure.

(d) Flexibility

By flexibility was meant the ability of the compressor to operate satisfactorily away from its design point so that the engine could run at reduced speed, accelerate without surging, and so on. This was an especially difficult subject, since compressors of a given type varied in this so much. Certainly if extremely high range of mass flow at a given speed were required, then the centrifugal was generally the best, but this in itself was not a useful requirement for gas turbine engines. It was the position of the characteristics at various speeds that mattered and the range of efficient operation of the outlet conditions. In this respect it might be shown that the axial compressor at the same compression ratio was not inferior to the centrifugal as was shown by Fig. 4, in which, although the range of the axial compressor was much less on inlet conditions, it possessed a similar range on outlet conditions.

He had also made a direct comparison of two actual compressors at the same compression ratio and scaled to the same flow in Fig. 5. Their relative sizes for the two types might be seen and their characteristics and jet engine working lines were also shown. Both characteristics were suitable for the engine. The axial might be a little stickier on acceleration at low speeds, while the centrifugal would be worse and might even surge under altitude conditions.

Either of these possibilities might be avoided by working rather farther down the characteristic, or by other ingenious devices. An eminently suitable compressor of either type was shown but the axial was much more efficient, had rather less than half the frontal area and considerably less bulk.

CONCLUSION

He would like to conclude with some remarks as to why he believed the axial compressor must develop more than the centrifugal, provided that a reasonable proportion of work was expended on both types.

Firstly, the axial compressor could be split up stage by stage, its separate performance analysed and even tested separately. The detail flow at all stages could be examined by well known methods. This possibility of step by step development had always been found a great advantage and was the reason why gas turbines had developed so rapidly, because the compressors, combustion chambers, and turbines were readily separated and examined on rigs. Further, it was possible without serious discrepancy to idealise the problems somewhat and make rig tests of the blading in the form of cascades of blades and this technique had enabled a thorough knowledge of the behaviour of axial compressor blading to be established. The complete application of this knowledge to the three dimension problem was by no means perfect, but great benefits to performance and efficiency had already been realised. No such analysis or breaking down of a centrifugal blower was really possible, as each part essentially demanded the presence of the others, for even an approximately correct functioning.

Secondly, he believed the axial compressor to be in the direct line of general aerodynamic development. All the concepts and methods of wing theory were readily carried over to axial compressor blade rows. The improvements in high speed wings from the study of boundary layers had their equivalent in blade theory and he believed that equal improvements were obtainable. No such condition existed in the centrifugal blower whose progress must rest on a large number of *ad hoc* tests, guided by good general aerodynamic knowledge, but no more.

He would leave to his colleague, Mr. Lovesey, the subjects of reliability and cost and the practical results being achieved with axial compressors, both on test and in flight.

THE CENTRIFUGAL COMPRESSOR — Mr. BRODIE

Nowadays, with large engine projects, it was practically impossible for any engine manufacturer to carry the full cost of design, manufacture, and development, from his own finance. For the adequate completion of such projects Government assistance was sought and obtained, as a general rule. It was probably true to say that such large engines could not be developed economically to production standards to satisfy purely the relatively small civil market and in practically all cases of which he had knowledge, the military outlet for these engines was predominant and was the prime reason for obtaining official financial support.

If an engine were designed and developed as part of a military weapon, its manufacture in time of war must receive first consideration. From this point of view there were quite serious criticisms to be levelled against the axial compressor. Its multiplicity of small, very accurately manufactured blades and discs made this unit expensive, from the point of view of man hours utilised. It would seem, when discussing the two types of engine, with all things being equal, it was only in the compressors themselves that such differences existed as to make one a *serious* production problem and the other just a production problem. He did not propose to discuss, therefore, any further section of these engines in this criticism.

In an obviously inspired paper in one of the national weekly aviation papers a closely reasoned statement called aptly "Two Edged Blades" suggested that if three quarters of all military types of aircraft were fitted with axial type engines, a monthly output of some 5½ million blades, of varying shapes and sizes, would have to be envisaged. There was little doubt that in terms of man hours of work, factory space, and capital equipment, this would mean a major national problem. Even half this number would be a headache.

Not a great deal was known in Great Britain about the really big production of axial flow engines, but he was assured by people in other countries who had experience of side-by-side manufacture of centrifugal and axial flow engines of equal

power output, that the axial was at least twice as expensive to produce in large quantities.

For some years detailed studies had been made of various forms of axial flow engines and production experts, covering an immense amount of ground, had spent a great deal of time in establishing costs from various methods of production. The close-to-size forging was today probably the most economical method of producing compressor blades and in this matter they were well advised on current practice in this country. For an axial of, say, twelve stages, by duplicating blade forms as much as possible and cropping them to suit their individual stages, at least sixteen sets of dies would be required per engine. Those dies cost approximately £500 to £600 each and the production of blades from them would be roughly as follows:—

Aluminium	3,000 per die
Aluminium Bronze	1,500 per die
High temperature steel	Between 300 and 500 blades per die

The cost of those forgings would vary between 5/- and £3 each. In addition, they would require machining on the root attachment and leading and trailing edges and tips, involving 0.25/0.50 man hours of work per blade. Their limits of accuracy would probably be of the order of 0.004 in. to 0.006 in. as an envelope of the blade form. He would leave them to work out the cost of producing 1,500/2,000 of those things per engine. It would seem that considerable increase in skilled labour staffs would be required in the Tool Room to manufacture dies, in addition to those required in the Machine Shops for final machining of blades. On the other hand the die cost for a centrifugal impeller would be approximately £2,500/£3,000 with a life of many hundreds of forgings which would cost approximately £100 each.

He thought it was true to say that axial compressor blading was manufactured in Great Britain generally on adapted standard machine tools and very little had yet been done to develop specialised machine tools capable of handling, with the necessary degree of accuracy, the phenomenal quantities required. Many interesting developments were afoot and he had little doubt that in the course of time much would be done by production and machine tool experts to narrow the gap in production time between the two types of engine, but this would take time, and in a country's pressing need during an emergency, time was at a premium.

It had been estimated that to produce an equivalent number of axial flow engines to centrifugal engines, approximately twice the factory floor area would be required for the former and nearly four times as many machine tools. Inspection of parts alone raised no mean problem as, over and above dimensional and weight checks, he had heard that in certain rather temperamental axial compressors it was necessary to do a vibration check on each blade prior to its assembly, in order to ensure that failures due to vibration fatigue would be reduced to a minimum. This check was made on finished machined blades and the scrap rate from it was reported to be high.

On the subject of serviceability he had no direct experience in the handling of axial flow engines and his information, therefore, was mostly second-hand. As Dr. Moult had already mentioned they had, over the years, ample proof of the magnificent digestive system of the single-sided centrifugal compressor. He had a deep certainty that had some of the material which, to his knowledge, had passed through this type of engine without serious damage, attempted to pass through any of the axial flow engines which he knew, massive damage would be the result in the latter. He considered that this question of vulnerability of the axial compressor was, and probably would remain, its greatest weakness. Retracting guards on air intakes might provide a palliative but were useless in combating the effect of ice or hard objects met in flight.

Another matter which he thought should be mentioned again was the loss of performance over extended periods between overhauls. It was known that the drop in performance due to the fouling of the compressor was higher in respect of the axial and he felt sure that the advantage of the type in fuel consumption would be progressively reduced due to this cause.

He was aware that methods were being developed for cleansing axial compressors after periods of running, but while such methods might cope with adhering dirt they did not remove the erosion of the blading surfaces which came after many hours of operation, and a decreasing return was obtained from each subsequent cleansing.

THE AXIAL COMPRESSOR — Mr. LOVESEY

They had heard an excellent analysis of the advantages of both types of compressor, but they were not called upon to make a decision on which type made the better engine based on this information alone because, in Great Britain they had in production and in service, engines with both types of compressor.

He proposed to review briefly how engines with axial compressors confirmed the claims put forward by Mr. Pearson, and how, in practice, the axial compressor stood up to some of the criticisms levelled against it.

The data he had was based on some 8,000 hours' running, of which 2,500 hours were in flight.

(a) Efficiency

It could not be disputed that the axial jet engine, at the present state of development, gave substantially better fuel consumption than the centrifugal engine under all conditions. By all conditions, he referred to very complete measurements taken in flight over a wide speed range and up to an altitude in excess of 50,000 ft. As an example he could quote two engines of somewhere near the same thrust, in which the axial engine gave 17 per cent. lower specific consumption. As an indication of what this meant, the difference in the measured amount of fuel used on a 150 hour development test under type test conditions was of the order of 13,000 gallons in favour of the axial type. It would be seen that it did not take long for the axial type to save on the fuel bill the difference in the initial cost between the two types.

(b) Flow per lb. frontal area

Mr. Pearson put forward the case of low frontal area and, in support of this, he could say that a test was made on a modern aeroplane with wing installations of both centrifugal and axial jet engines in which the nacelles were tailored to the respective types. The range of the aircraft was reduced by at least 20 per cent. because of the difference between the specific consumption and the increased drag. He knew the critics would say that a number of other factors should be taken into consideration, but all they amounted to were excuses for the centrifugal type, and he did not know of a single case where excuses had to be offered to justify the axial engine.

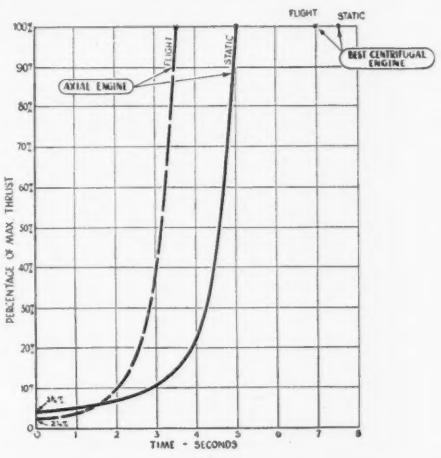


Fig. 1.
Acceleration characteristics of an axial jet engine.

(c) Weight and compactness

For this aspect he would give an example of the two types of engine which had reached the same standard of development, *i.e.* capable of running a satisfactory 150 hour

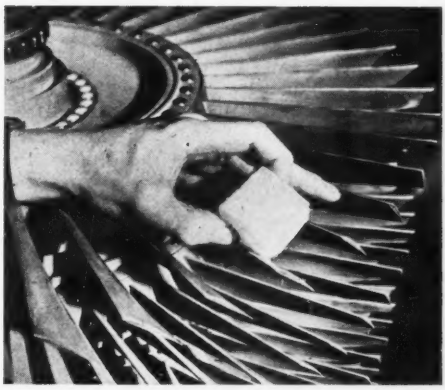


Fig. 2.
Avon 5 icing tests, showing size of ice cubes (1½ in.) passed through the compressor, relative to the first stage rotor blades.

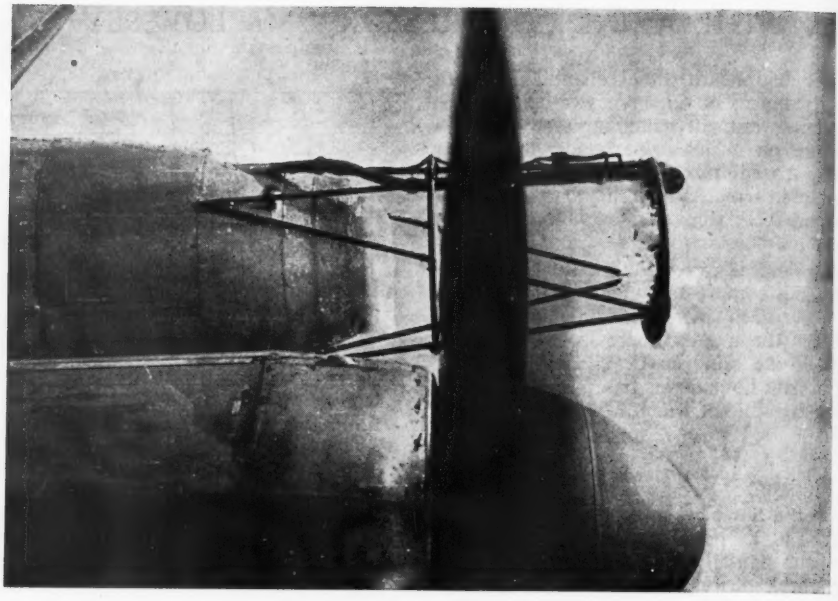


Fig. 3.
Icing tests on an axial engine in a Lancastrian.

test under type test conditions. Both gave exactly the same specific weight figure of 0.282 lb. weight/per lb. of thrust, and both engines operated at the same flame temperature. One important difference was that the axial gave 70 per cent. greater thrust per sq. in. of frontal area than the centrifugal.

(*d*) *Flexibility*

There was a general impression that axial engines were less flexible from the pilot's point of view than the centrifugal. This was probably true in a number of cases but they were making axial engines that had better handling qualities, both on the ground and in flight, than any centrifugal engine that was flying today.

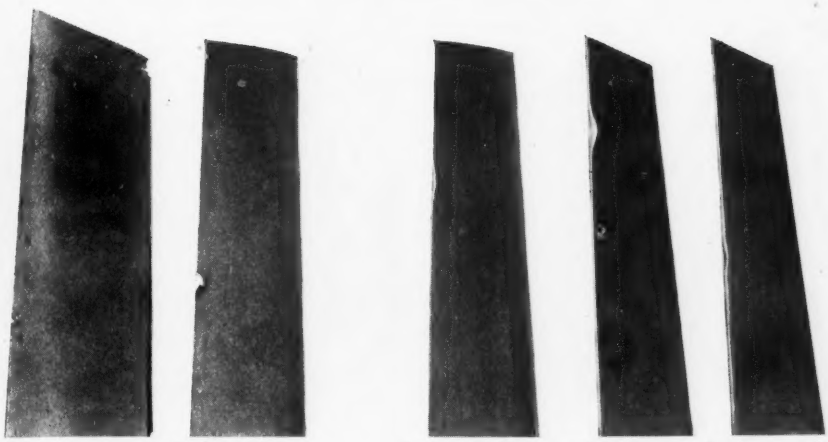
Acceleration times, from idling to maximum thrust, of five seconds were being obtained on the ground, and between three and four seconds in flight.

It had not been an easy task to achieve this result, but the criticism of inferior handling properties could not now be used against an axial engine.

(*e*) *Reliability*

Experience obtained during 8,000 hours development running of an axial jet engine, including some 2,500 hours in flight, showed that generally speaking the reliability was just as high, if not higher, than on the centrifugal engine.

Fig. 4.
Ice build up in axial compressor. Conditions:— −10°C., water concentration 0.6 gm./cu. metre, 12,500 ft., 325 m.p.h.



Showing damage before reclaiming — Reclaimed blades, satisfactory for further running

Fig. 5.

Axial compressor rotating blades. Damaged blades.

There was a lot of evidence that the axial engine was very much kinder to the combustion equipment than the centrifugal, due, no doubt, to the smoother air flow into the combustion chamber than was obtained from the centrifugal, which had to turn the air through an angle of 90° or more.

It had been said that the axial must be much more vulnerable to the effects of ice; but this had not been the case, and very early in the development they had made some drastic tests of feeding 1½ in. cubes of ice through the compressor, as shown in Fig. 2. No damage at all resulted, and the compressor blades were not even marked.

Following this, icing conditions were simulated in flight on a Lancastrian by spraying water into the intake, as shown in Fig. 3.

During those tests ice frequently became detached from the spray grid, but no damage resulted.

Continued application of the water spray did result in a build up of ice on the inlet stator vanes, as shown in Fig. 4, which resulted in a loss of thrust up to 20 per cent.

Conditions tended to stabilise due to the ice becoming detached from the stator vanes, again causing no damage to the engine. Naturally, some form of de-icing was necessary in order to maintain thrust, but the experiments made so far tended to show that the axial engine was nowhere near as vulnerable to damage from this source as was originally anticipated.

It was difficult to assess the relative vulnerability of each type of engine due to the entry of foreign bodies other than ice. He could only quote cases where they had had the steel mandrels used for fitting hollow head rivets go through the compressor. This did result in damage to the blades, but the damage had not been such that it prevented

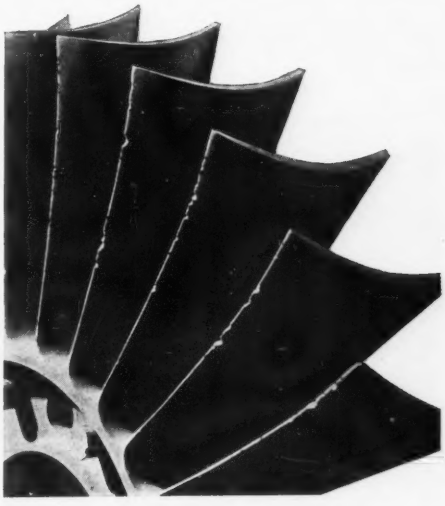


Fig. 6.

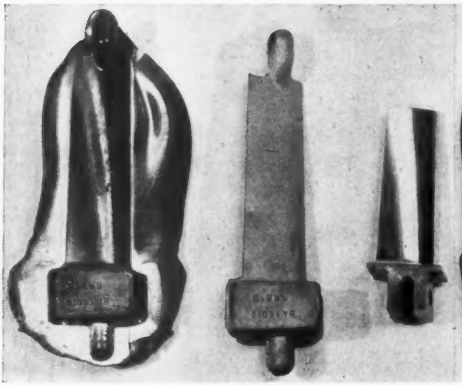


Fig. 7.

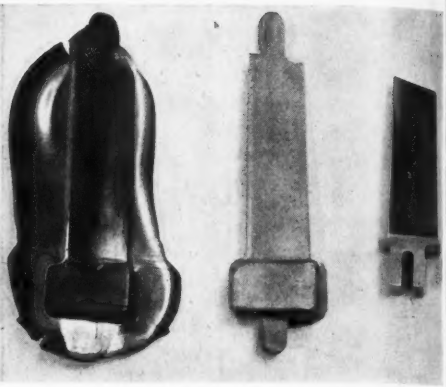


Fig. 8.

the engine operating satisfactorily for long periods. In fact when such damage had occurred it had not been found until one of the periodical inspection periods of the engine. The sort of damage caused to the blades was shown in Fig. 5, and it had been the practice to re-condition engines by filing out the damage marks.

Similar damage resulted to the inducer vanes on centrifugal compressors, as shown in Fig. 6.

With the entry of large pieces of steel, or stones, he would guess the chances of either type in surviving would be more or less equal.

(*f*) *Cost*

He had left this item to the last because he knew very little about it, which was probably only a little less than anyone else. There was no question that the axial type engine, in the development stage, was infinitely more costly than the centrifugal type. At the present moment production engines were still considerably dearer to produce in the axial form, and compressor blades were the chief reason for the higher cost.

Very rapid strides were being made in the technique of blade manufacture, and the stage had now been reached where blades were pressed to size and the only machining

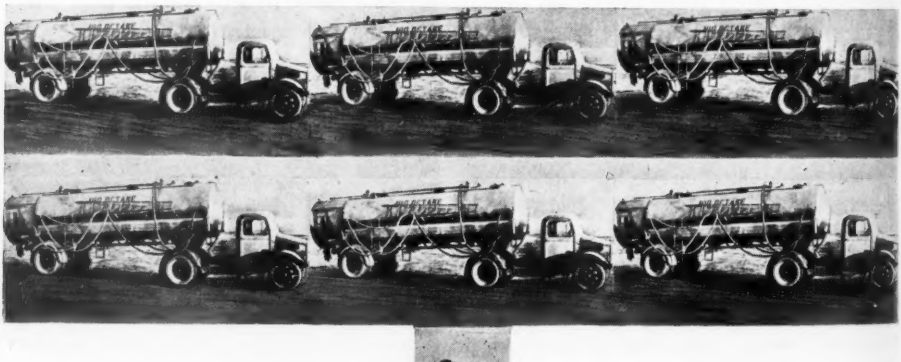


Fig. 9.

operation was on the root fixing and trimming the leading and trailing edges.

The only way to cheapen the engine was to get committed to production and the cost was bound to decrease with the numbers produced. He thought that the axial would always be a little dearer per lb. of thrust than the centrifugal type.

Figure 9 represented approximately the extra amount of fuel used by one centrifugal engine in 500 hours' flying on typical bomber operation, due to the difference in specific consumption alone. When considering cost, this important item must be taken into consideration.

If in the future, it should be proved that centrifugal engines were the better type then all he could say was that they had enjoyed their experience with the axials, but, if it should be proved that the axial was the better and their opponents came forward with an axial engine, then he would feel that they deserved some credit for converting the unbelievers.

GENERAL DISCUSSION

Major F. B. Halford (de Havilland Engine Co. Ltd., *Fellow*): The discussion had carried his mind back about ten years, when axial blowers had existed and they had the choice to go to either the centrifugal or the axial. They were then at war; the axial was suggested, but they felt that it would take too long to develop. Five years later, in 1945, when the war had ended, they were told that no more centrifugals were required, and that the axial was the engine of the future. Today, five years later still, in the Royal Air Force there were no axial engines! Dr. Moult and he had had to go to America to find out something about them in Service use; it would have been nice to have stayed at home and to have looked at such simple and un-costly jobs, which had been developed in so short a time, but they could not do that! If an official war were to break out tomorrow it would be fought largely with the centrifugal engine throughout the world. His greatest fear was that they might not be able to overcome the problems of excessive man hours in manufacture; in America the place was being turned upside down in order to overcome the difficulties of the cost of the axial-type engines.

With regard to the provision of ice guards, bullet guards, and so on, the digestion of the axial was not so good as the digestion of the centrifugal.

If, as had been said, it had been simple ten years ago to develop the axials and to get results, where were they in the Royal Air Force? One would like the explanation! There were several thousand flying in America. On one occasion he had asked a Chief Engineer what was the specific consumption of axials now in use in America, and was told that the lowest figure was 1.05. It was always "jam tomorrow," and the argument came down to the centrifugal of today compared with the axial of tomorrow. He wanted to see the axial of tomorrow, but he wanted to see it cheap and easy to make! He felt sure that in the next five or ten years that would be achieved; but it had not been done at the moment.

In the "starting" figure he could not decide how to compare the centrifugal with the axial, for the running up had started from idling! Presumably the engines were always kept running, day and night?

While their palates anticipated the jam of tomorrow, it was wise not to neglect the bread and butter of today.

F. M. Owner (Chief Engineer, Engine Division, Bristol Aeroplane Co. Ltd., *Fellow*): He had been privileged to share in the development of two engines, using combined axial and centrifugal compressors. The axial could digest large quantities of air but needed many stages to produce high pressure ratios; the centrifugal, on the other hand, could give quite high ratios per stage, but could not deal with such large quantities of air as could the axial.

It was rather extraordinary that in the discussion so far there had been so much insistence on what they had now. For good or ill, the die that was cast five years ago was resulting in what they had now. But the purpose of such debates presumably was to enlighten them as to where they should go next.

As one who read the American press no less assiduously than the British, it was becoming the practice to refer to output in

terms of five figures instead of four. Again, the space available in aircraft was becoming steadily less. It seemed to him, therefore, that they could not by any means ignore the paramount importance of frontal area.

The answer to digestive ability was that they must conduct the bolts, overseers' overcoats and similar indigestible material elsewhere; but must make the compressor reasonably tolerant.

The factors of increasing speed, output and compression ratio compelled them to go to the axial type in the future, whether they liked it or not; and whatever the difficulties in compounding and other developments in the axial to give the characteristics that they must have, they must be faced and solved.

W. H. Lindsey (Armstrong-Siddeley Ltd., *Fellow*): He was convinced when he came to the meeting that the thing to do was to make axial flow compressors; there had been a lot of side-tracking on the subject of the centrifugal, but the fact remained that, if in the future they wished to carry heavier loads farther and faster than they were today, there was no other way of doing it than to resort to the use of the axial flow compressor. It gave a smaller frontal area than the centrifugal and therefore reduced drag, it gave higher efficiency and it could be made to give higher compression ratio; and all those factors added up to lower fuel consumption.

It must be agreed that they did not know much about operating experience with axial flow engines. But, as Mr. Owner had pointed out, the purpose of the debate was to try to make up their minds on what they should be doing now, and not to decide whether or not the course taken five years ago was then the right one. There was even a hint of agreement in favour of axials in the remarks made by Mr. Brodie, for it seemed that his contribution could be summed up as meaning, in a sentence, that axials were the better compressors, but they could not afford them.

L. J. Cheshire (English Electric Co. Ltd.): Some years had passed since he was connected with the Aircraft Industry, and although at that time he had been fully occupied with the development of the centrifugal engine, he had been equally fully occupied since in developing the industrial axial compressor.

It seemed a pity that the discussion had been so controversial, because all the participants were doing their best in the interests of the country to develop the particular types of engines in which they believed. Some serious decisions would soon have to be made at a high level; and while noting the arguments which had been put forward on both sides during the discussion, he could not help recalling that for very many years the "experts" had been pro-axial. He believed it was correct to say that Whittle's first proposals were turned down by two Ministries. Fortunately Whittle had had a little money and was very obstinate; in those days it was possible to be obstinate for a little money, whereas now it would cost a very great deal and the Ministries were better organised! Who would say that Whittle was wrong? The astonishing and rapid success of the centrifugal engine had given the axial compressor people plenty of time to make their engine what it had recently become; in spite of what had been said by its advocates, it was only comparatively recently that the axial engine could equal the centrifugal even on specific consumption, but it was undoubtedly true that the axial engines now being made represented remarkable

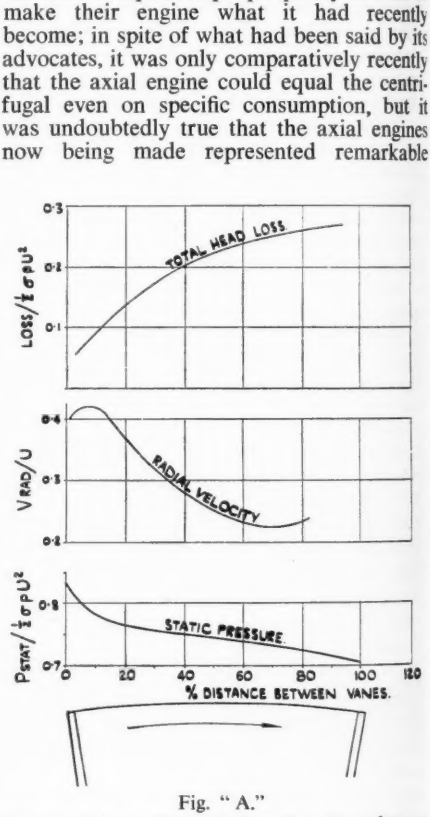


Fig. "A."
Low speed tests of flow in impeller channels near the tip.

achievements. He felt that the problem should be considered rather from the point of view of the duties the engines had to perform. There were many occasions when frontal area was of vital importance, but there were others when it was not.

The centrifugal engine had many virtues, and there was no question that it was simple to produce. The problem was not merely that of choosing between a good engine and a bad one; it was a question of having available one type of engine or another, or both, and if it happened that by the choice the capacity of Great Britain to produce engines were seriously jeopardised, it became of great significance, so that he did not think the development of the centrifugal compressor ought to be dropped.

Aerodynamicists had always seemed to be afraid of the centrifugal compressor problem, the general nature of which could be fairly simply stated.

Figure A* illustrated the air flow through a centrifugal impeller, to indicate the essence of the aerodynamic problem involved. The air came through the intake, up through the impeller channel and out of the impeller into the diffuser system. If the impeller were to rotate in the direction shown by the arrow, the pressure on the leading face of the vane should be higher and the pressure on the trailing face should be lower; and since there should be equal energy at any given radius, that should result in the radial velocity at the leading face being lower than at the trailing face. Similarly, the total head loss should be constant. But what happened in fact was very different. There was a radical departure from the condition of lower radial velocity at the leading face. Those tests, made by Mr. A. G. Smith on a low speed rig, showed that the flow through the channels of centrifugal impellers was very disordered with regions of high static and total head loss; and indicated the direction in which they should go to cure the trouble with those compressors.

The reasons why the centrifugal compressor was at present not so efficient as it might be were thus quite well known, and he felt that the virtues of the centrifugal compressor — its simplicity, its undoubted reliability and its attractions in other ways— were such as to warrant paying significant attention to its one major defect, *i.e.* its lower efficiency. More attention should be paid to the aerodynamics of the centrifugal compressor; it had been neglected consistently by aerodynamicists in Great Britain and in America, but from the point of view of their needs in time of war, if from no other, it should be tackled, because once they had improved that efficiency they would have made a reliable jet engine, or propeller engine, which would be at least as efficient as the axial and would have many other attractions.

No reasons had been given so far why the centrifugal should have a lower efficiency than the axial. If the centrifugal compressor were inherently of lower efficiency than the axial, then the aerodynamicists should tell them why, and they would give up hoping. But they did not tell them, nor did they tackle the problem. The problem was extremely difficult and the work would be expensive, but a lot of money had been spent on axials.

Hayne Constant (Director, National Gas Turbine Establishment, *Fellow*): He had worked for several years on centrifugal compressors, but about 1936 he had seen the light, and the last thing he would like to do was to start on the centrifugal all over again, as suggested by Mr. Cheshire. He felt that they had made a big mistake in splitting their forces between the two different mechanisms; that mistake had since been rectified, and he hoped that no one would start it again.

It had been a little surprising to hear the confession by Major Halford, who had looked around the country and the world and had said how ill prepared they were for war. He had alleged that they had no axials capable of going into the Royal Air Force yet, and that if there were a war now they would have to fight with centrifugals. He had also said that ten years ago he had been asked to make axials and had refused, and that five years ago he was told that the axial was the engine of the future, but he still went on making centrifugals.

For some years past quite a lot of people had advocated the use of axial compressors by Great Britain, and he was glad that that advocacy was now rapidly making itself felt.

On the technical side, in the comparison of axial and centrifugal compressors it seemed

*Reproduced from "Design and Development of Centrifugal Compressors for Aircraft Gas Turbines," L. J. Cheshire, *Proceedings of the Institution of Mech. E.*, 1945, Volume 153.

that people had tended to be too greedy in claiming all the virtues for the axial simultaneously. Most people would admit that the axial engine was more efficient than the centrifugal. An advantage of 17 per cent. had been mentioned by two speakers in the discussion; from all the data at his disposal he had deduced that the axial engine was 15 per cent. more efficient than the centrifugal. It might be that American axials were not much more efficient than the American centrifugals; but the best figures that could be obtained from a piece of mechanism, and not the worst, were always used when making comparisons.

In respect of frontal area he believed everyone would agree that the ratio was roughly 2/1, in spite of the fact that Dr. Moult had produced an outline diagram which purported to show that, for a centrifugal and an axial engine of the same output, the frontal areas were much the same. He would have liked to have seen inside that diagram!

He was surprised that it had not been said that the axial engine was as light as the centrifugal, for today it was indeed, at the same performance as the centrifugal, lighter.

By using the axial compressor they could achieve either lower weight, or better fuel consumption, or less frontal area, or lower costs. But they could not expect to achieve all those advantages simultaneously; they could probably achieve two or three of them simultaneously, and equality in the case of the remainder. He thought that that was good enough.

E. L. Carlisle (Air Registration Board): As an impartial observer he could see both the engine man's side and the airline operator's side; and the axial flow compressor might be described as a race horse and the centrifugal as a shire horse. In the former case they had a delicate animal which had to be carefully nursed and pampered; and if it could be started, and if the conditions were right, it would give a good performance for a short time. On the other hand, the shire horse was a sturdy, robust animal, which would slog away regularly at heavy tasks under all conditions, needing a minimum of attention and with no fall off in performance.

They had seen the result of leaving a small nut in the air intake of an axial, when a shovel had to be used to deal with the broken blades. That nut could be a stone picked up on take-off, or a hail stone, or a piece of ice encountered in flight; and they had yet to prove how successful de-icing was. They had also to see the effect of the stronger material used for the blades of the first two stages in the axial. The clearances were so small that it was doubtful whether or not it would be the absolute cure. But the centrifugal type of compressor could swallow small objects without doing expensive harm; indeed, it could swallow airmen's caps without any bother. They had seen the adverse effect on performance of particles of dirt adhering to the blades of the axial, and he wondered what would be the effect of sand on those delicate structures. Its rival had none of those inhibitions.

The airline operator would have to make up his mind whether or not he was going to spend his money on expensive machines, entailing large quantities of expensive spares which would need elaborate maintenance, and special handling as a power plant and aircraft on the airfield, for some slight improvement of performance; or on a robust engine which needed a minimum amount of attention, had no expensive expendable items, and which required no special handling, but would give regular performance in the hurly burly of airline service.

As a private individual and not as a member of the Air Registration Board, he believed that, all utilisation being equal, the type of engine which Dr. Moult had advocated would attain 1,000 hours' overhaul life long before the axial.

H. Roxbee Cox (Past-President, *Fellow*): Nearly all the speakers so far were practising designers and knew far more about the technical points than he did, but he felt that from his point of view "on the sidelines," the matter was perhaps not one for controversy at all! There certainly was a tendency to divide into two sides, and there seemed to be a cleavage, not only in Great Britain but throughout the world, between axials and centrifugals. Instead of there being a division on the matter, it might be more reasonable to suppose that there was a place both for the centrifugals and for the axials.

It would be wrong to assume that the whole trend of development would be always towards bigger and bigger engines and hence, *vide* Mr. Owner, axial compressors, or towards engines for military purposes.

There might be a case for building relatively small engines for quite a time to come; furthermore, presumably there would be a place for civil aeroplanes. Those who put forward the virtues of robustness and reliability had a great deal on their side, and it seemed that for certain purposes the centrifugal compressor engine must continue for a long time to have a place. He would not wish to pour too much oil on the troubled waters, because while they were disturbed they would probably get more instruction out of them than otherwise. But it might be wise to admit that there was room for both and that they would indeed be wrong in the future to neglect the development of the centrifugal engine. He sympathised with Mr. Constant's implication that there were not enough facilities to enable them to do everything; but he believed there were sufficient to enable them to continue the work on the centrifugal, and he hoped with Mr. Cheshire that some further effort would be made to improve a type of compressor which he believed would pay for being improved.

G. R. Edwards (Chief Designer, Vickers-Armstrongs Ltd., *Fellow*): He had the somewhat difficult task of deciding which, among the various combination and permutations of engines, was the one best suited at any particular time to put into an aeroplane. He would really like to continue the "oil pouring" which Dr. Roxbee Cox had started and say that there was probably a place for both axials and centrifugals for some time to come.

It had been said that the stage at which an engine became a good civil aircraft engine was when it was obsolete; maybe that was in accord with Mr. Carlisle's comment that the centrifugal type of engine might well continue to be a good civil aircraft engine for a long time; he felt that some had been much attracted to the centrifugal engine for civil aircraft purposes because they knew that it was reliable and robust. One of the reasons why the development of the centrifugal engine had been carried much farther during the past ten years than had the development of the axial, was surely that the centrifugal compressor in a turbine was a logical development of the centrifugal blower which had appeared in so many piston engines. So that when they put a centrifugal compressor turbine into an aeroplane they were putting in a piece of equipment which had been used in civil aircraft for a long time with the knowledge that it had a certain reliability, whereas they did not know how reliable was the axial. He would have thought that the reliability of the axial compressor, particularly from the point of view of vulnerability, must also be a function of its size; stones and pieces of ice were of reasonably constant dimensions, and surely the resistance of a blade to the impact of such things of constant dimensions must be dependent to some extent on its size. The demonstration by Mr. Lovesey of the way in which ice and mandrels had passed through an axial compressor was a most convincing demonstration of the ruggedness of the axial compressor of the size discussed.

The point made by Mr. Constant must be fundamental, that they never achieved all the advantages all at once and they must assess the values of the different types and decide which was the best for the particular job in hand. If they wanted a high performance aeroplane which had to travel a long distance, they had to make it small in order to give it high performance; that reduced the room available for installing engines and, much more important, it reduced the amount of room available for carrying fuels; it might be a physical impossibility to design an aeroplane of specific size to travel a certain distance unless engines were available of proved specific consumption. Events during the past year had tended rather to resolve themselves into, so far as military aeroplanes were concerned, a competition between one axial and another, rather than between an axial and a centrifugal.

They ought to derive a great deal of comfort from the fact that they had a constellation of engine designers in Great Britain who had had sufficient foresight to ensure that there were a lot of centrifugal engines available at this time, the production of which could be expanded very quickly in the event of war at short notice; and additionally they had some advanced engines which, as Mr. Constant had inferred, were probably a good deal better than some of the others that were somewhere else, and would be very useful for any war they might have to fight a little later on.

Miss A. Kennedy (*Companion*): With the fluctuating availability of fuel which they seemed to be experiencing, and judging by the demonstration Mr. Lovesey had given of the extra fuel consumption of the centrifugal

type of engine as compared with the axial, the extended use of the centrifugal might create quite a major problem.

V. N. Butler (Omes Ltd., *Associate*): He was interested in the manufacture of blades and had worked under Mr. Brodie many years ago. In Great Britain they had now reached a production of 1,000 stainless steel blades per die, so that they were a little in advance of the condition Mr. Brodie had stated and perhaps could be a little more optimistic in respect of the production of the axial compressor. It must be said in all fairness that the forging people had not really been put to the test so far as the axial compressor was concerned; they had never had to produce the very large quantities of blades envisaged and had not been encouraged yet to go to the high degree of mechanisation that might be necessary in the future. But they were just as much concerned as were the engine designers and manufacturers in the performance and production of blades, and quite a number of forging people in Great Britain were "bending their elbows" in that connection.

G. L. Wilde (Rolls-Royce Ltd., *Assoc. Fellow*): Mr. Cheshire had said that it was difficult to appreciate why the centrifugal compressor efficiency should be fundamentally lower than that of the axial compressor; surely, this was due to the high Mach numbers existing in the centrifugal machines used in jet propulsion engines. In order to obtain 4 : 1 pressure ratio in one stage there was no alternative but to put up with the high relative velocities which were now common practice, and herein lay one of the chief obstacles to raising the efficiency. Fundamentally, therefore, the comparison often drawn between centrifugal and axial compressors was made under conditions unfavourable to the centrifugal type.

A relative Mach number of 0.9 onto the impeller intake guide vanes was common, and at the impeller tip relative to the stationary diffuser vanes values of over 1.0 were reached. Under those conditions not more than 70 per cent. of the kinetic energy at the impeller tip could be converted to pressure, and this fact was the chief reason for the lower efficiency of the centrifugal compressor. A pitot tube at the impeller tip gave efficiencies of 85/88 per cent. If they restricted themselves to 2 : 1 pressure ratio, single-stage centrifugal units could be made to attain 86 per cent. overall efficiency, but the corresponding axial would be higher. With regard to axial diffusers applied to centrifugal compressors, it was his experience that anything from 5 to 10 per cent. lower efficiency was obtained.

Some effort was being put into developing the supersonic axial compressor, but so far only efficiencies of the rotor had been quoted. Although those were high (in the region of 85 per cent.) nobody yet seemed to have put a row of stator blades behind the rotor and to have obtained an overall figure. This was badly needed, and it would then be interesting to see how the figure of overall efficiency compared with that for the centrifugal compressor.

MR. PEARSON'S REPLY

He could claim a little credit for having admitted at the beginning of his discourse that there was a case at the moment for both types of engines. Indeed, his Company were making both, and they were not anxious to get rid of the centrifugal just yet, for quite a lot of money was involved in its development. But he maintained that for the future the axial compressor held out more scope for development.

For some years all the engine manufacturers in Great Britain who had been working on jet engines had prided themselves on being in advance of the Americans; that was true in respect of axials as well as centrifugals, and yet **Major Halford,** in order to find out, must go to America. Also, he appeared to have been misguided with regard to Reynolds number when in America, because quite different information had been imparted to others from the same source. Tests, made by Rolls-Royce Ltd. under altitude conditions, seemed to point to the virtue of the axial compressor in that respect; but he did not claim that much at the present stage because, naturally, it was desired to check such things many times. Obviously much depended on the scale factor, on the particular size of engine used. The case which Major Halford and others had put for the centrifugal was largely that if they had a war immediately they would need centrifugals; and it was certain that his firm, as well as the Rolls-Royce Company, would help. But it was a pity if Major Halford's case needed a declaration of war to support it!

Mr. Cheshire apparently thought that somebody high up in the Ministry of Supply had a difficult decision to make soon, but the decision had already been made.

The Comet had been mentioned. He had heard something terrible about it, that it would suffer the infliction of the axial engine! Perhaps it would then be a better aeroplane!

Mr. Carlisle put his money on the shire horse. He had also referred to the damage caused by nuts in axial engines; but nuts were more troublesome in development than in service. Some trouble had been taken to overcome that difficulty in connection with axials, and it would be shown eventually that the axial was just as reliable in service as the centrifugal. Again, it was largely a question of size, and he believed that, when axials were made sufficiently large, they would be as reliable as the centrifugals.

A reason why centrifugals were relatively inefficient had been given by **Mr. Wilde,** that they were spoken of at compression ratios of up to only 4 : 1. He had deliberately shown a figure covering a wide range of compression ratios, from 2 : 1 to 7 : 1 for the two types of engine, and throughout that range there was a consistent difference of about 7 per cent. between the best axial and the best centrifugal in respect of efficiency, the axial being the more efficient. It was possible to show worse results with bad axials and bad centrifugals.

They had the promise that the axial engine would be more efficient than the centrifugal, smaller in frontal area, lighter and smaller in bulk, just as flexible and, taking everything into consideration, including the seven oil trailer cars, less costly. Therefore, for the future he claimed that the axial compressor would reign supreme.

DR. MOULT'S REPLY

He could not attempt to deal with all the points that had been raised; that would require a long time, but he hoped that all who considered the problem would be swayed by deeds and not by words. It seemed from Mr. Pearson's last remarks that the number of these oil trailer cars had increased by one in the course of the evening! Mr. Lovesey had shown a picture of six cars representing the extra amount of fuel consumed by a centrifugal engine in 500 hours of flying. It was doubtful whether certain axials would last that long!

Mr. Pearson had said that the Rolls-Royce Company were constructing both types, and that was rather an admission. The de Havilland Company had had to make a decision many years ago, and he was quite sure they had taken the right decision at the time. They could not afford to back the horse both ways, and subsequent events had fully justified what was done.

The discussion had been encouraging. It was certainly encouraging on his side because both Mr. Cheshire and Dr. Roxbee Cox particularly, had said that the centrifugal had still some way to go, with which he agreed heartily; and it was encouraging also to the axial people, because at long last it appeared that they were getting somewhere!

He counselled discretion in connection with the figures that had been put forward—the axial people were still talking of the "jam tomorrow." For example, one curve, Fig. 2, had shown an axial having a mass flow of 30 lb. per second per sq. ft. of frontal area. Perhaps he was not fully informed on such matters, but in his ignorance he had thought that the current engines had almost achieved half that! The particular engine he had illustrated in his own diagram had a flow intensity of 14 lb. per second per sq. ft. of frontal area, and he did not think that was doing so badly. Incidentally, he would reassure Mr. Constant that it was intended that the diagrammatic centrifugal engine should have internal fittings.

In a recent paper Mr. R. N. Dorey (who was not unknown in Derby!) had said that their experience had shown that part of the price paid for the improved specific fuel consumption of the axial was a loss in handling flexibility and a general complication of the control mechanism.

As to Mr. Pearson's reference to the Comet, the aircraft was designed some years ago to fulfil certain anticipated duties. Those duties it fulfilled admirably but in addition there remained the longest and most severe stages of world operation, which could now be tackled with a larger engine.

As to the ability of engines to swallow debris, he would not attempt to compete in regard to the sizes of the lumps of ice, the wing spread of birds, or anything else which might be drawn into the engines. But the point had come out during his visit to America that in the course of ground attacks, as when rockets were bursting on ground

targets, the engines might collect a lot of debris. Whatever had been said about the merits of the two types of engine, the centrifugal undoubtedly stood a better chance of survival than the unprotected axial in similar circumstances.

Mr. Cheshire had felt that the advocates of the two types of compressor had been unduly contentious. He would be prepared to apologise about that, but it was their job to be controversial on that occasion. If they had brought to light some of the pros and cons of the two types they had achieved what they had set out to do.

The President

In closing the discussion he would not endeavour to deliver a verdict. The speakers compared, in their capacity for marshalling their facts, and possibly in drawing red herrings across the paths of their opponents, with eminent Counsel to whom he had listened recently in the course of his duties as Foreman of a jury. The speakers in the discussion were deliberately invited to be controversial, for it was by honest and hard-hitting argument that they elicited facts and helped one another to come to conclusions. They could claim that British jet engines were still the best in the world, and largely because there was not just one single channel of endeavour; there was no idea that there was only one right way. It was their British practice, and always had been, that people who had faith in their own line should pursue it and achieve success, and one man, by his efforts, acted as a spur to another. God forbid that they should ever decide that there was only one "right way," for he felt sure that if they did that they would sink.

He had hoped that more detailed consideration would have been given to the problems of the relative production of the two types. He understood that the present cost of an axial compressor unit might be taken broadly as twice the cost of a centrifugal compressor gas turbine engine. Reference had been made to a remark, originally made, he believed, by Lord Tedder, "that in war one cannot afford a second best," but there was not necessarily one "best" for all purposes at any one time; not every aircraft needed was either a fighter or long range bomber in which sheer performance was the outstanding criterion. "Good" engines for those other purposes which could be produced in the quantities required would be more useful than rather "better" designs which could not be provided at the critical time. In his capacity as President of the Society he felt bound to stress the vital importance of the factor of production in the selection of engines to which an expansion programme in emergency was committed. The utmost attention must be given not only to "efficiency" shown on paper, or even confirmed by a few prototypes, but also to the ways and means by which such engines could be created within the time limit imposed and with the resources which could be made available.

METAL ADHESIVE PROCESSES*

by

F. H. PARKER

INTRODUCTION

Metal adhesives have been known to the Aircraft Industry for at least ten years, the Chrysler Cycle Weld process being among the earliest. Since then metal-to-metal bonding has been the subject of many papers, but no member of a firm of aircraft constructors is known to have lectured on the subject.

It is hoped, therefore, that this paper may prove a useful contribution to the workshop aspects of the method. The fundamentals of adhesion will be avoided and only brief reference will be made to the strengths of structures assembled by that means; those are matters for the physicist and aircraft designer. The process and manufacturing technique will be considered together with safeguards and controls as used in the Industry on metal aircraft construction.

The Bristol Aeroplane Company first took an active interest in metal bonding at the beginning of 1948. A start was made on available materials which were fully approved under D.T.D. 775 and the combined use of these materials is known as the "Redux" Process. This first step was made, and no reason has since been found for change, by using the combination of "Redux" liquid K.6 and powder conforming to 30-52 screen, B.S. 410-1943 and, except where otherwise stated, these are the materials discussed. Experiments over the past three years have disposed of many doubts about the reliability of bonded structures for aircraft and what is, and is not, important has now been satisfactorily determined.

The steps taken to reach this satisfactory position are described.

*A Section Lecture given on the 28th November 1950—the 810th Lecture to be read before the Royal Aeronautical Society.

Mr. Parker is Development Engineer, The Bristol Aeroplane Co. Ltd.

CURING EQUIPMENT

THE MINIATURE PRESS

The first press was quite miniature. It consisted of a laboratory moulding press, with a platen size 7 in. × 7 in. This equipment proved useful in reaching first approximations but a rather larger pilot plant is advised, capable of curing ten specimens simultaneously as a more representative number. This press should have all the mechanical and heating settings capable of being accurately pre-set and controlled, thus allowing a methodical check on the several factors outside its mechanical operation, for example control of glue thickness, exact repetition of loading, heating cycle, and so on.

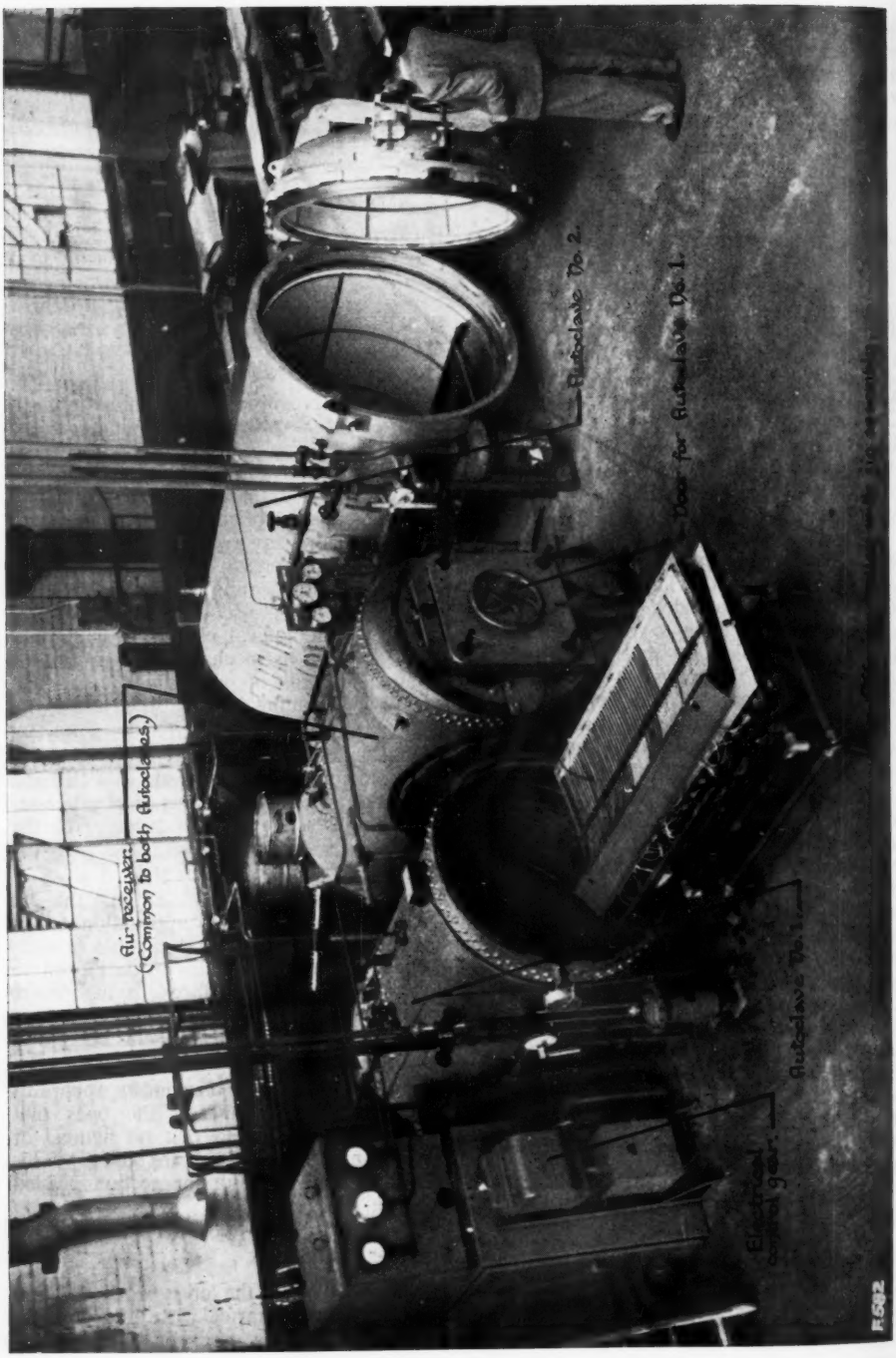
The standard shear strength figure of 1,500 lb. (D.T.D. 775) was accepted as a minimum for the selected resin and the first task was to try out the sequence of processes on a small scale in accordance with the manufacturer's instructions, using first the press curing technique.

At the outset it was realised that the flat platen press had a limited application, and it was decided that a method capable of wide application would need to be developed to cover, for example, surfaces having severe double curvature. This led, through stages to be described, to the present successful "Autoclave" method.

In the description of elementary apparatus already referred to and in the ones that follow, it will be noted that no figures for pressure and temperature are given. This subject is dealt with in the section headed "Curing."

FIRST PRESSURE BOX

Concurrent with the press work the first experiments were started with fluid pressure curing. A box was used consisting of two


Air receiver.
(Common to both Autoclaves.)
Autoclave No. 2.
Door for Autoclave No. 1.
Electrical control gear.
Autoclave No. 1.
F.6832

chambers capable of being clamped together one on top of the other. The lower chamber was used as a steam cavity to heat the work specimen. The space in the upper chamber was loaded with first a metal die conforming to the shape of the part to be formed, then the component; on the top of this, and restrained by the walls of the chamber, was a rubber bag which, when pressurised from a supply, gave the required pressure on the work specimen.

With increase in size this equipment would become more and more unwieldy, while the rubber bag would be expensive and easily damaged. This crude start established the principle of fluid curing and the necessity for fitting accurately male and female dies, with the attendant great cost, was avoided while uniformity of pressure all over the component was assured.

The autoclave

The next obvious step was to abandon a system in which the air pressure was applied unilaterally and to evolve a bilateral method in which it provided its own reaction. This was done by immersing the parts to be bonded in compressed air, taking special care that no air could get between the faces to be joined together so that this cavity remained at atmospheric pressure. Thus the autoclave process was evolved.

A miniature test autoclave was made from an 8 in. diameter steam pipe 18 in. long, together with a special type of electrically heated jig, and a process was evolved which proved satisfactory and capable of being reproduced immediately afterwards to a larger size without further difficulties. Thus in one step, with practically no change of method, the full-scale equipment was established.

Figure 1 shows the full-size equipment. Reading from left to right is the first full-size autoclave with door open and a jig ready for entry. The next unit is a high pressure air accumulator which, using only a small capacity compressor, allows a very quick charge to be made to the autoclave; to the right is a larger autoclave also charged in sequence from the same accumulator.

Figure 2 gives an idea of the arrangement of the jig within the autoclave, and Fig. 3 illustrates the make-up and operation of the jig. This consists of a light (14 standard wire gauge) aluminium alloy plate conforming to the final shape of the component, attached to a number of cradle frames to hold it to form. Immediately under the jig plate, woven electric elements are fitted to provide the necessary heat.

The component to be cured is laid on top of the jig and this in turn is covered with a thin 1/8 in. rubber sheet which is clamped

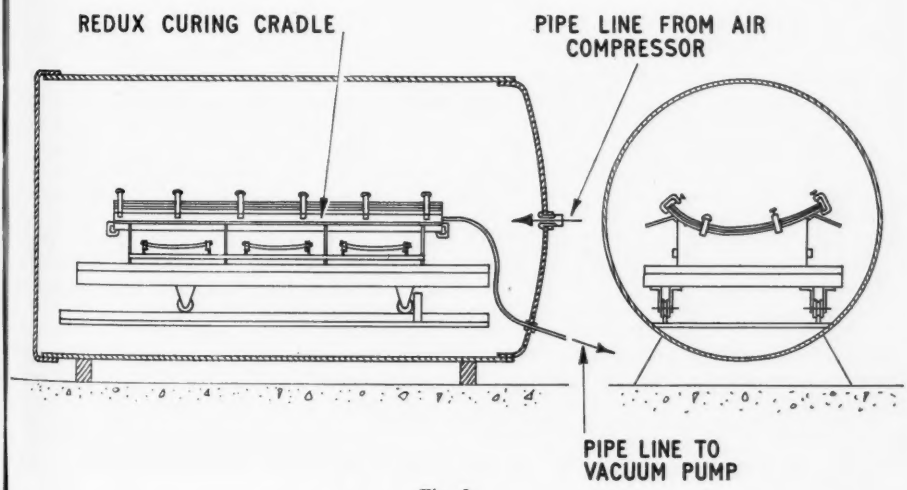


Fig. 2.
Sectioned view of curing chamber.

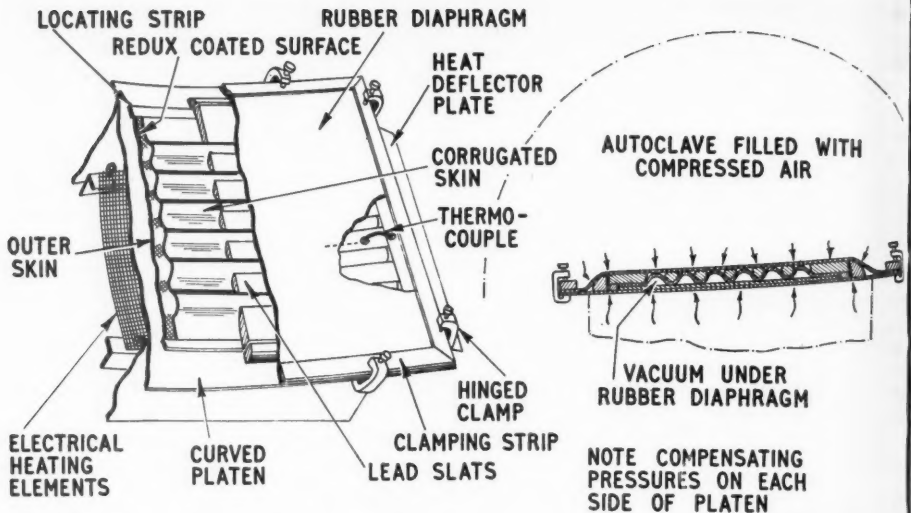


Fig. 3.
Autoclave curing jig assembly and principle of pressure application.

down around its edges. A vacuum line is attached to the base of the jig communicating with the cavity between jig and rubber blanket. The sequence of operation is as follows:—

The jig is loaded as described and electrical and vacuum connections are made outside the autoclave. Vacuum is then applied; this evacuates the rubber blanket and in addition completes the edge sealing of the rubber, thus creating the required differential pressure across component and jig, which on being confined in the autoclave can be brought up to any required curing pressure.

With the form of structure illustrated, in calculating the required curing pressure on the joint, consideration must be given to the proportion of surface being bonded to the total area under uniformly distributed air load, since the glue line area provides the sole reaction for the whole of this force; thus with the typical corrugated structure shown, where the glue area is approximately 1/3 of the total area under pressure, only 60 lb./in.^{2} pressure is required in the autoclave to give a bonding pressure on the joints of 180 lb./in.^{2}.

It should be noted that all electrical connections to the supply source are made automatically when the curing jig cradle is passed into the autoclave and all the necessary safety precautions, both with regard to compressed air supply and electrical supply, are incorporated in the door-closing mechanism.

The platen press

While the autoclave process was being thoroughly established, the platen press was not neglected. At present a 250 ton press fitted with 2 in. thick precision steam-heated platens 6 ft. × 3 ft. is in use. This is shown in Fig. 4 and consists of a four-post, double-ram, rising-bottom platen machine. The steam heated platens are mounted on one in. thick insulating material and work is fed into and out of them from feed tables on either side of the machine. All controls, including a curing cycle time-warning clock, are conveniently grouped, so that the operator has all the controls and the component under observation.

Two larger presses are about to be installed, one being a long narrow open-face press which will be used for bonding edge reinforcements to skin panels.

PREPARATION FOR CURING

The two stages leading to the application of heat and pressure are:—

(*a*) Pre-treatment of the metals to be joined, and

(*b*) The application of the liquid and powder adhesive.

PRE-TREATMENT

From the outset it was known that the pre-treatment stage was of vital importance, but experience has shown that the control is not so exacting once production is under way. Use is always made of pickle to D.T.D. 915 A. even if this is subsequently followed by anodising to D.T.D. 910 B. It has been found in addition to vapour de-greasing that the use of an aqueous de-greaser ("Collex") is also advisable. Aqueous de-greasing effectively removes any adhering solids frequently left by vapour de-greasing.

With the exception of the vapour de-greasing which is done at any convenient point between the raw material store and detail manufacturing stage, all the pre-treatment plant should be installed in the metal bonding department and its use should be in the care of those responsible for all other stages of the process. Careful record must be kept of the operating conditions, *e.g.* time, temperature, bath concentration, and so on.

The sequence of the operations and the compositions of each bath used is given in Table I. The plant is shown in Fig. 5.

During early tests both on the press and in the autoclave, pre-treatment was suspect for every failure. Water impurities were exhaustively examined. Results following different methods of drying were tabulated, as were the results following ordered variations in bath concentrations. In short, all the variables were systematically explored with the result that a method of pre-treatment was standarised, the application of which requires only a slightly more exact shop control than is necessary for a normal plating shop, but the bonding plant and

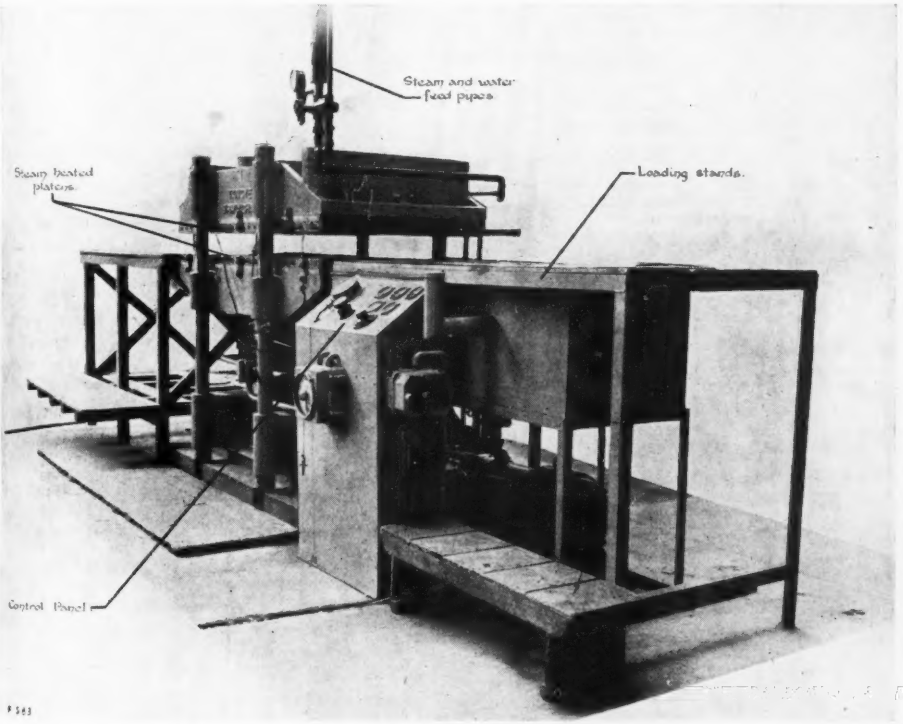
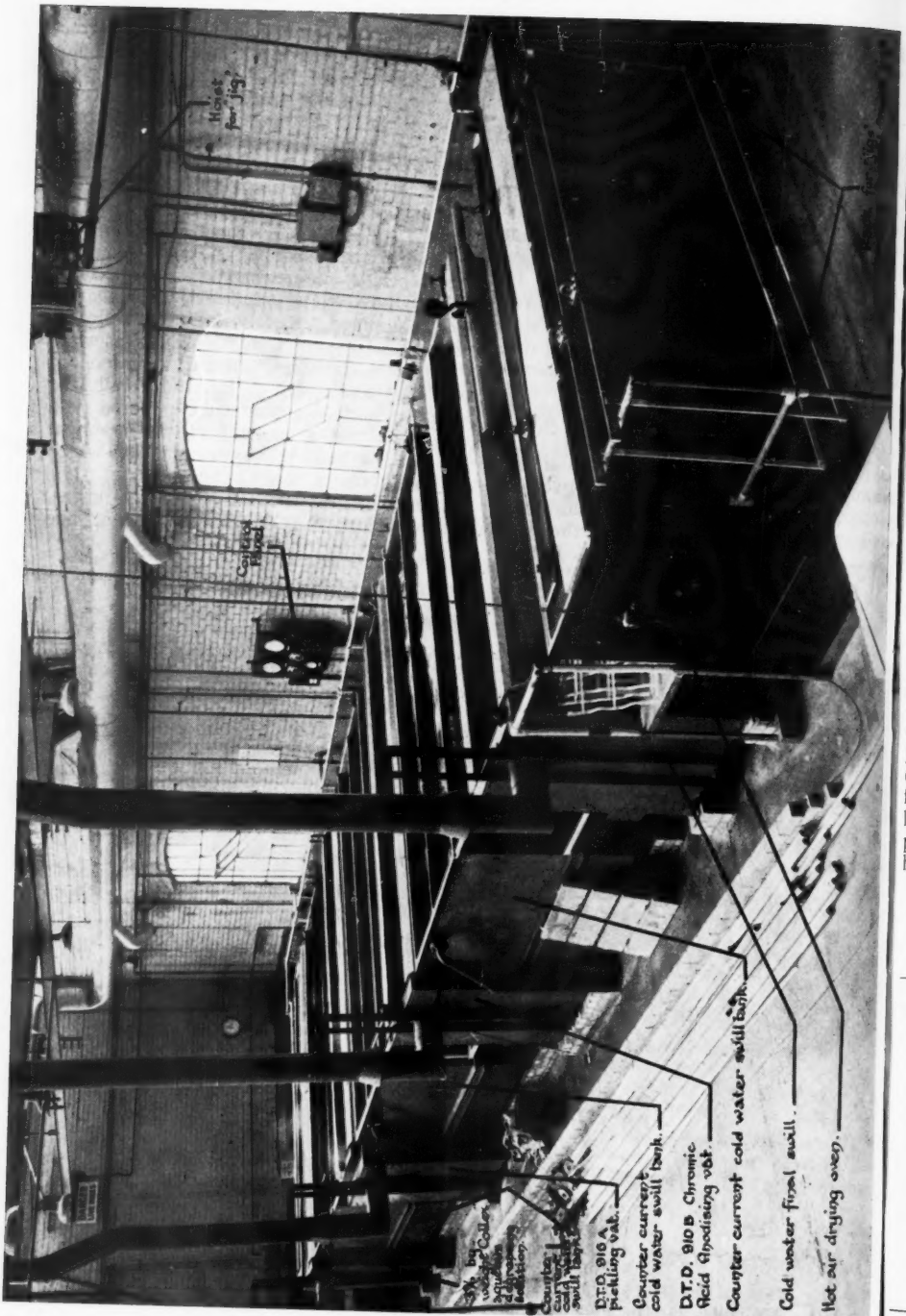


Fig. 4.
Hydraulic flat-platen "Redux" Curing Press (The Bristol Aeroplane Co. Ltd.).


Hoist for jigs
Cat-walk Panel
Counter current cold water swill tank.
D.T.D. 910 A Pickling vat.
Counter current cold water swill tank.
D.T.D. 910 B Chromic Acid Anodising vat.
Counter current cold water swill tank.
Cold water final swill.
Hot air drying oven.

pre-treatment baths should be completely separate from any other operation.

All these investigations will be found recorded in Ministry of Supply Scientific and Technical Memorandum No. 7/49, while a further S. & T.M. to be issued shortly should provide a full picture of this vastly important phase in the sequence of processes. The memorandum quoted contains many references to the full subject.

Application of the Resin

From pre-treatment and after drying, the prepared material is taken over by the process operator for resin application. The bonding operator handling the job at this stage is responsible for ensuring that bonding faces are not contaminated. The method of resin coating is that of brush application of liquid K. 6 and subsequent dip application of the powder; the residue after consolidation on the resin face is lightly shaken off. Mating parts are then either wired together or rack stored until ready for curing.

Curing

The curing process has received much examination and it was evident throughout development of the press and autoclave that despite identical pre-treatment and subsequent resin application, the appearance of joints made on the press was noticeably different from those cured in the autoclave; furthermore, the autoclave joint always had higher properties, particularly in regard to peel strength. The joint when separated by hand shows much more uniform and ductile properties. At first this was thought to be associated with uniformity of contact resulting from air pressure curing, as well as with other factors present.

On both systems the curing pressure was 200 lb./in.^{2} for 15 to 20 minutes at a temperature from 145°C. up to 150°C. maximum with an open assembly time, *i.e.* period between resin application and curing, of about one hour. This short interval was introduced to avoid shop contamination.

Press curing under these conditions resulted in peel strengths of great irregularity and the visual appearance of broken joints was unsatisfactory. The press technique, however, could not be abandoned as it had some advantages and offered a slightly shorter cycle, while jigging is normally cheap and simple for flat assemblies, for example, for many sheet-stiffener combinations.

Pre-treatment was re-explored and every mechanical variation in the press and heating equipment was perfected. The outcome was the use of a longer open assembly time and a reduced curing pressure at 100 lb./in.^{2}, for use on the press method.

Uniformity of pressure and even temperatures over the full area of the parts to be joined are essentials to successful curing; thus the platens must fit accurately at closure and the steam passages through them must be sufficiently numerous and so disposed that the temperature is truly uniform.

The essential of a good joint is strict control of all factors which influence the flow of the adhesive. Excessive flow causes undue loss of adhesive from the joint, often

TABLE I

Bath sequence composition and operating conditions of Redux Pre-treatment Plant

Sequence	*Bath*	*Composition*	*Operating Conditions*	
			Time	*Temperature*
1	"Collex" de-greaser	3 per cent. by weight "Collex" aqueous degreasing solution	20 min.	80°C.
2	Swill (double)	Filton bore hole water	5 min.	Ambient
3	Pickle	Pickle to D.T.D. 915 A. (5 per cent. by weight chromic acid, 15 per cent. by volume sulphuric acid)	Min. 20 min. Max. 30 min.	60°C.
4	Swill (double)	Filton bore hole water	5 min.	Ambient
5	Anodic	Chromic acid anodise to D.T.D. 910 B. (3 per cent. by weight chromic acid)	1 hr.	40°C.
6	Swill (double)	Filton bore hole water	5 min.	Ambient
7	Swill	West Gloucestershire Supply water	Dip	Ambient
8	Hot air oven		30 min.	Approx. 50°C.

accompanied by a disturbance of the proportion of resin to powder. It is necessary to control temperature rise and pressure, and to ensure evaporation of all solvents before the joint is closed. In the autoclave this condition is met, since the pressure is applied at a uniform and compensating rate and thus the resin is partially cured and consolidated before the full pressure is reached. On the press, however, harsh mechanical crushing can occur at the beginning of the cure.

Thus the introduction of autoclave curing and lower pressure press curing, on the whole, has increased the overall strength of Redux bonded joints while scatter has greatly reduced. This has resulted in higher average strengths than were obtained originally. Average values are 2,200 lb. shear on a joint 0.5 in. × 1 in. and 60 lb. peel.

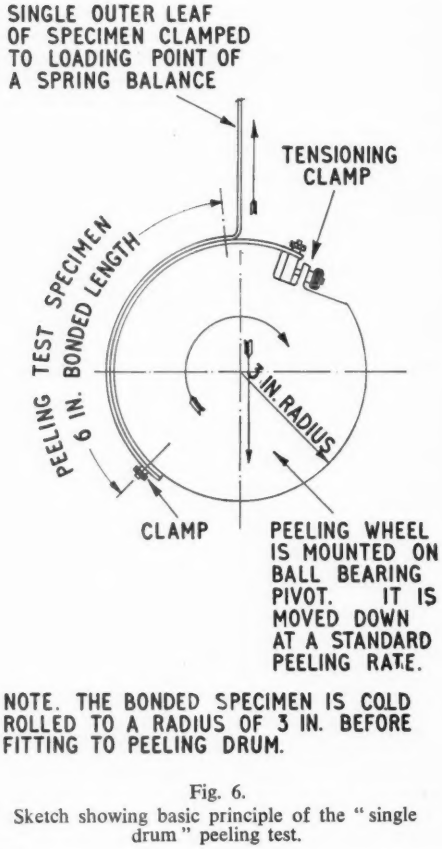

SINGLE OUTER LEAF OF SPECIMEN CLAMPED TO LOADING POINT OF A SPRING BALANCE
TENSIONING CLAMP
PEELING TEST SPECIMEN 6 IN. BONDED LENGTH
3 IN. RADIUS
CLAMP
PEELING WHEEL IS MOUNTED ON BALL BEARING PIVOT. IT IS MOVED DOWN AT A STANDARD PEELING RATE.
NOTE. THE BONDED SPECIMEN IS COLD ROLLED TO A RADIUS OF 3 IN. BEFORE FITTING TO PEELING DRUM.

Fig. 6.
Sketch showing basic principle of the "single drum" peeling test.

Press pressure may be still further reduced, provided that the component being cured does not itself require considerable force to bring the mating parts into uniform contact.

With the autoclave working on thin gauge material excellent bonds have been obtained using atmospheric pressure only; that is, the pressure has been that arising from the use of the vacuum between the cradle and the rubber sheet.

PEEL TEST

The peeling test was developed by Aero Research Ltd., and has been established as the main control test in the absence of any known form of non-destructive examination of the bond.

The test consists of peeling against an autographically recording loading device. Fig. 6 shows in diagrammatic form the method of applying load to the specimen.

It may be seen that the peel specimen is clamped to a circular drum; the free end is held in a suitable clamp. The drum is then pulled hydraulically, which causes the specimen to rotate and peel the two component strips apart.

As a result of thousands of tests a minimum peeling load of 30 lb. has been established on a specimen made from 20 gauge D.T.D. 610. In this connection lies the chief limitation of the single drum test. If the test figures are to have any significance for comparative purposes, the test must be confined to one gauge and material specification since, by nature of the geometry of the test, any change in the gauge of material or specification alters completely the peeling strength.

A further disadvantage is the fact that the actual radius which the specimen takes up as it leaves the drum, varies with stiffness of the material and increases the loading moment on a specimen of low strength, while decreasing it on a specimen of high strength; thus scatter of the results is so severe as to make them meaningless; the need for standardisation of gauge and specification of material is essential when making comparisons.

Means are being developed to eliminate these limitations and a peel test will soon be available which will allow immediate intercomparison of test results on materials of different gauges and specifications up to a certain limit, when the peel test becomes virtually impossible. But even with these

limitations the single drum peel test is an excellent means of checking the process, since it has been found that it shows up immediately any slight variation in pre-treatment or other bonding conditions and it is regarded as one of the most valuable checks on the quality of a joint. It is interesting to note that the Aeronautical Inspection Directorate are likely to establish this test as a standard requirement for accepting Redux bonded joints. It is not suggested that the peel test has any significance as a design criterion, unless perhaps it is associated with endurance against fatigue.

On the other hand, it is well known that the shear strength of Redux bonded joints is very insensitive to processing conditions. There is abundant evidence of joints with no apparent peel strength easily meeting the shear strength requirements of D.T.D. 775.

INSPECTION PROCEDURE

Inspection procedure is a difficulty with metal bonding since there is no non-destructive test which can be applied intelligently and evidence of quality is limited to a test on a separate specimen.

It has been found necessary, therefore, to introduce strict inspection routine to each stage of the method, and although at first sight this appears to call for a high ratio of inspection to production labour, that is amply met by the low overall manning required.

A metal bonding plant producing, say, 1,000 components of different design and sizes a month requires initially a production manning of about 25 operators, with 6 to 8 inspectors in addition. As the operators become experienced and if the rules are properly followed the inspection manning can be reduced safely by half; even so the inspection ratio is higher than for the older methods of assembly, but the overall manning is very much less.

Inspection control consists of a stage by stage viewing throughout the several processes. The inspector's last dual jobs are the shear and peel tests which in each case are on one test piece per component per pre-treatment batch. A typical peel graph is shown in Fig. 7; each graph is signed by the inspector. The back of each graph, which is seen on the left hand side of the picture, also carries the routine check records which the inspector has made. Thus the completed slip, stamped by the inspector, gives a permanent record of the process history of the part to which it applies.

DESIGN AND WORKS CO-ORDINATION

With metal bonding the closest design, development and production collaboration is essential. The design must be appropriate for the method from its inception; its adoption merely as an alternative to riveting is often no advantage. The essential requirements for good bonding must be realised by the designers and catered for in component layout.

Apart from overall considerations, such as the obvious one of panel sizes to fit autoclaves, there are a number of considerations such as the strip reinforcement of thin sheet edges by bonding. This permits the use of fewer rivets at panel edges, which also makes possible cut-countersinking for final assembly with its attendant economies. Edges thus thickened permit great weight economies over the main panel area while allowing loads to be taken through the reinforced edges. This feature is well illustrated in Fig. 8.

Local load concentrations can easily be accommodated through laminations secured by the metal bonding which, in some cases, is particularly useful where stringers and rib members terminate on thin panels.

SOME REDUX BONDED STRUCTURES

Two important applications were apparent in the initial design of the Bristol Type 175.

(*i*) The wing leading edge reinforced with an internal corrugation, the multiple channels thus formed acting as ducts for hot air de-icing purposes.

(*ii*) The assembly of large inter-spar ribs, particularly where smooth surfaces are requisite to prevent chafing on flexible petrol tanks.

These applications are shown in Figs. 9 and 10.

Another valuable application lies in the use of thin materials with reinforcing members attached so as to ensure a large bonding area, avoiding stress concentrations and thus minimising fatigue possibilities.

SERIAL NO. 149
DRG. NO. H20303 NDS
M.O. 26463
SEQUENCE 1/695
NO. OFF 1 off.
DESCRIPTION
Redux S.A. Fwd.
Diaphragm F.O.

JOB HISTORY

STAGE 1 PRETREATMENT

TYPE	Pickle & Anodise
TIME OUT	1.30 p.m.
DATE	17.5.50

STAGE 2 REDUX APPLICATION

DATE	17.5.50
TIME	3.30 p.m.

STAGE 3 CURING

PLANT	Autoclave
DATE	18.5.50
TIME IN	8.50 a.m.
TIME OUT	9.10 a.m.
TEMP.	150°C.
PRESSURE	150 lb □"
VACUUM	21 Mercury Ins.

STAGE 4 FINAL INSPECTION

ACCEPTED BY	
DATE	22.5.50

Shear Test :- 2,095 lb

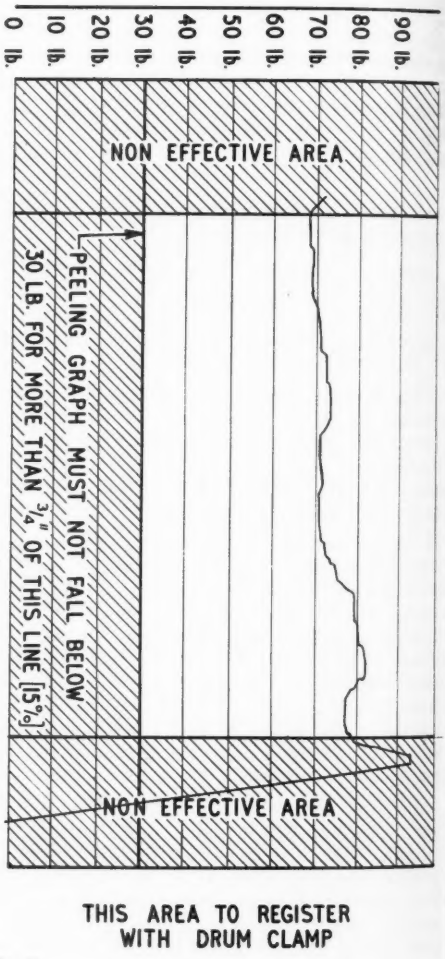


Fig. 7.
Face and reverse of standard peel test graph form.

ADVANTAGES OF METAL-TO-METAL BONDING

The use of metal bonding has four main advantages:—

(*i*) Reduction in structural weight

(*ii*) An improvement of surface finish

(*iii*) The reduction of stress concentrations

(*iv*) Reductions in cost.

It would be wrong to assume that all advantages follow automatically on the adoption of the method. For example, if minimum structural weight is the first objective, thin highly stressed surfaces are a necessity which, in many cases means closely reinforced thin sheets; but no means have yet been evolved of producing such structures without very perceptible external waviness. Thus if smoothness is the first consideration then some sacrifice in weight must follow. There is no difficulty in getting relatively thick (say 18 s.w.g.) sheet bonded to stiffeners without any visible surface deformation, but this may be achieved only at the expense of some structure weight. For large aircraft with high wing loading these conditions should easily be fulfilled, but in other cases a choice one way or the other must be made.

COSTS

Discussion on this important subject is rather premature, but some relevant factors may be mentioned. For comprehensive application of the method, initial capital cost of the plant is an important item. On the other hand, maintenance and replacement charges should be less than those incurred by the older methods of fabrication for metal aircraft. Designing for the use of the method should lead to simpler constructions than

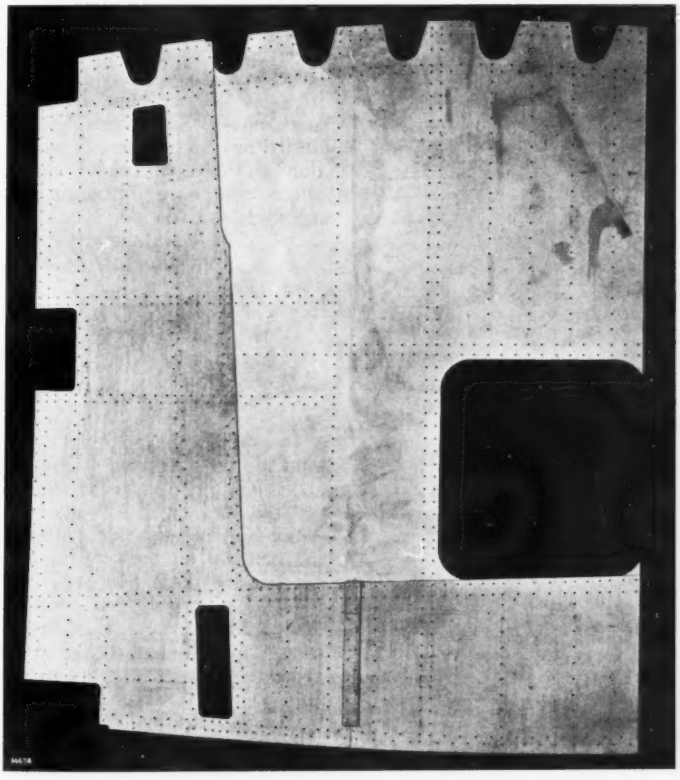


Fig. 8.
Wing rib diaphragm, edge-reinforced by " Redux " bonded laminations.

hitherto; there should be fewer parts and the components on the whole should be lighter. All this should show to advantage in the final cost account. Designers must realise that the costs associated with curing time are as large for a small component as for a large one, and that, within reason, whatever complications are built into the product the cost of curing is not affected. Once in routine production reductions in cost should follow the adoption of the method.

Profit-raising of aircraft operation which should follow the realisation of greater disposable load follows in the wake of the weight saving that should accrue. Examples of weight saving are those which must inevitably arise from the reinforcing of sheet edges leaving the main body of sheets of the desired original thickness and similarly, from the economic end reinforcing of stringers, which are known to involve a large saving in weight.

SOME RESULTS

Large numbers of panels in a variety of sizes, from the small laboratory specimen to full-scale aircraft panels, have been tested both for static and fatigue strength.

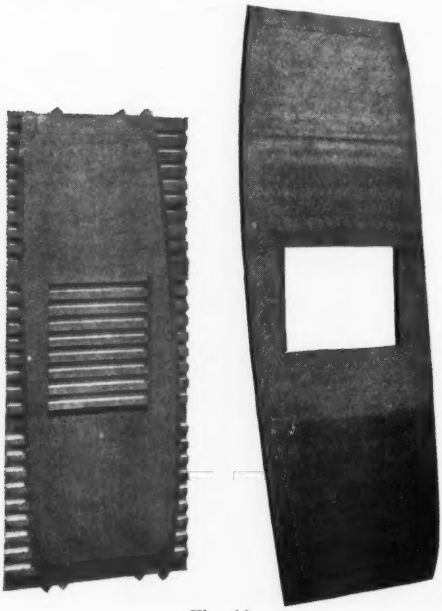


Fig. 10.
" Redux " Bonded inter-spar wing ribs
(30 s.w.g. metal throughout).

Fig. 9.
Section of wing leading edge assembly.

It has been found that shear and compression tests made on Redux assembled panels give results at least equal to similar riveted structures and in many cases the load carried is 30 to 40 per cent. in excess of riveted assemblies. Sometimes an original riveted design has been copied in which the distribution and size of panel elements has not necessarily been optimum for a bonded assembly. Even so, such assemblies have shown some advantages.

Typical figures for compressive failing loads are:—

Bonded	*Riveted*
44,000 lb.	42,000 lb.
37,000 lb.	34,000 lb.
53,000 lb.	50,000 lb.

It would appear that the principal factor contributing to added strength with stringer-sheet combinations is continuity of contact or the avoidance of inter-rivet buckling, which can be a source of premature failure in riveted structures. In some cases where the riveting pitch has been rather wide, the strength difference has been most marked. As would be expected, when a bonded panel is progressively loaded, failure develops from some region of instability but parting of the Redux joint is never the criterion and does not occur until final collapse of the specimen is imminent.

A major advantage of the use of metal bonding lies in reducing centres of stress concentration, and there is scope with intelligent

layout to add considerably to the fatigue life of many structures.

A good example of this is given in Fig. 11 which shows a helicopter skin with bonded stringers and local rivet reinforcings required to attach the skin to the rib structure. This part failed at a riveted joint, after 150 million cycles.

Two of the fundamental conditions for Redux curing are the application of pressure and heat; thus panel construction is obligatory and at this stage of development the contiguous panels have to be connected at their edges by riveting and/or bolting. Careful attention is necessary in the design of these joints if full advantage of the bonds is to be obtained, *i.e.* the efficient transfer of load from one edge to another.

No question relating to Redux has received so much publicity as that of the fall in strength with rise in temperature. The wing leading edge shown in Fig. 9 was tested at room temperature and also at 100°C. through the use of infra-red lamps. It is known that Redux loses two-thirds of its shear strength at 100°C., yet the two structures failed at nearly identical loads, appreciably above the design load. In neither case did the Redux bond fail. The reason is that in this, as indeed in most cases, the bond is subject to low intensities of stress but that may not be the case always and the problem of low strength at elevated temperature certainly needs solving at an early date. The increase in pealing strength which accompanies plasticity under heat is not the answer.

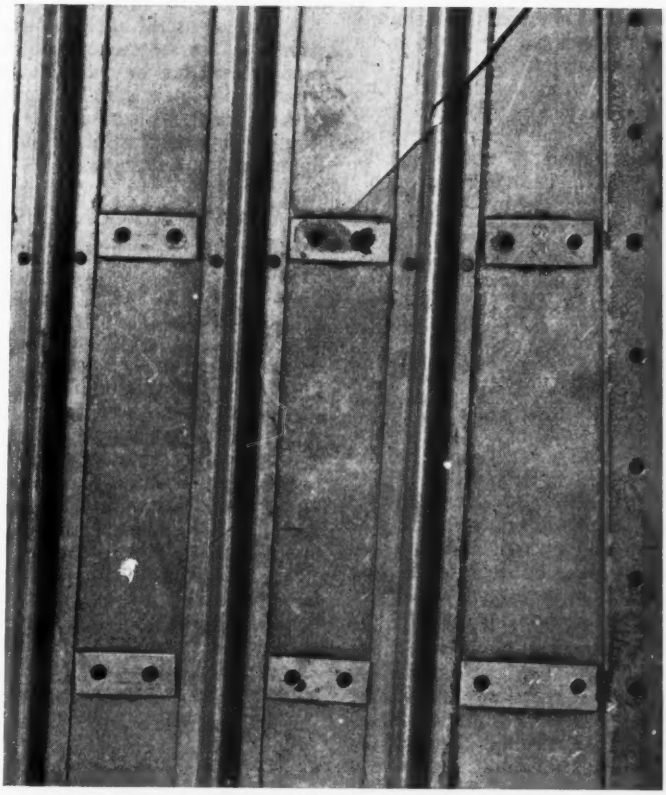


Fig. 11.
Fatigue failure in " Redux " Bonded skin panel
(skin 26 s.w.g, stringer 30 s.w.g).
Note undisturbed bond where crack traverses stringer.

OTHER BONDING MATERIALS

Many other bonding materials are available and have been considered. Among these are Araldite, also made by Aero Research Ltd., Hydunite, Cycle Weld (the Glenn Martin adhesive known as F.M. 45), Scotch Weld, Convair Metabond and a variety of Bostik derivatives. It has been found that most of the single part adhesives are extremely good adhesives when bonding relatively thick materials but are brittle and, therefore, may not be too safe in structures which are flexible; and fatigue life would be suspect.

If an attempt is made to bond together two thin materials, none of these adhesives seems to have any appreciable peeling strength and a parting, once started, rapidly runs the whole length of the joint.

Some adhesives, Araldite for example, have shear strength as good as, if not slightly better than, Redux, being over 2,200 lb. on a standard single shear test joint $\frac{1}{2}$ in. $\times$ 1 in. Tests made by the Bristol Aeroplane Co. Ltd. on the American adhesives have not been conclusive, due to shortage of supplies, but so far the shear strengths obtained have varied between 1,000 and 1,400 lb. or roughly half the value of Redux; but again there seemed to be a lack of peel strength.

The Bostik and Boscoprene family, because of their more plastic nature, exhibit promise of high peel strength but so far their shear strength appears to be low, ranging from 400 to 1,000 lb. on a standard single shear specimen. Also, on adhesives of rubber or similar base, evidence of joint "creep" at comparatively low stresses has been found.

MATERIALS BONDED

Tests have been made on the quality of Redux bonds on a number of materials and satisfactory joints have been produced with all of the following:—

Foremost are the aluminium alloys used in aircraft construction; all bond well within their elastic range. Then the magnesium alloys, including magnesium zirconium; steels, including stainless; rubber; asbestos based materials, for example Ferrodo or Ferobestos; Durestos; Tufnol and similar phenolic impregnated materials. There is no trouble with any of them provided that the appropriate and adequate precautions are taken at the pre-treatment stage.

Redux also gives a satisfactory bond between metal and glass cloth laminates made up with the Marco range of resins.

The relatively new and interesting material, magnesium zirconium, is likely to find considerable use in the Aircraft Industry. It is a metal on which, in the light of present knowledge, special attention should be placed on the avoidance of stress concentration. Thus metal bonding is an admirable process for fabrication and has further advantages in the matter of subsequent protective treatment within the joint. The best pre-treatment of the surface prior to bonding is still a matter for exploration, but it is anticipated that a really satisfactory chemical pre-treatment will be found, and that this material will be bonded with sufficient joint strength to permit the manufacture of structures of considerable economic attraction.

A further useful application is to bond a paper or fabric face to metal which, when prepared in this way, can be cold glued to wood or any other suitable surface, although with paper it is necessary to sand it away leaving only a film of Redux to develop optimum joint strength.

SUMMARY AND CONCLUSIONS

It would be misleading to convey the impression that by reading the available literature on the subject the desired results can be obtained immediately the plant is installed. There is no royal road to success in this matter and the best state of mind for anyone at the outset is that of a pioneer. On the other hand, there is no great mystery about the process and the fact that the efforts of those who have persevered have been rewarded by success should, in itself, give ample confidence and encouragement.

As already indicated, the first requirement is to install a pilot plant, including a pre-treatment pilot plant with vats sufficiently large to remove it somewhat from the sphere of the chemical laboratory, introducing something of the "shop element," but not so large as to make early and frequent changes in compositions uneconomic. Baths of about 50 gallons capacity have proved a good compromise.

The pilot plant should be set up in the department in which the full-scale work will be done, and should be put in charge of the carefully selected nucleus of personnel with whom it is finally intended to run the job.

This is important since the idea of sticking metals together does not come easily to people with long experience of the hitherto almost universal methods of metal jointing. It needs as long an association as possible with the process before everyone concerned acquires the necessary confidence in his allotted task. The early successful use of the Redux bonding process by the de Havilland Aircraft Company was in no small measure due to their long background of experience of gluing practice on wooden aircraft and their transition from wood to metal adhesion and finally, metal to metal, without the very definite break in adhesive methods which everyone else experienced.

ACKNOWLEDGMENTS

The author would like to express his appreciation to The Bristol Aeroplane Company Ltd., for permission to give this lecture, to the technical staff of Aero Research Ltd., for their help in the developments made, and particularly to his colleague, Mr. Hulbert, who has done a great deal of the basic work on the project.

DISCUSSION*

Mr. N. E. Rowe (British European Airways, *Fellow*), *contributed*: It was not clear what was being gained from the new method of joining metals together. From Mr. Parker's account there seemed to be little advantage in either man power or cost. He had demonstrated that smooth surfaces could be obtained, that reinforcement of thin plating could be done conveniently and that there might be advantage in fatigue resistance. Presumably the great advantage came from the saving in structure weight. What was the saving in structure weight expected on the Bristol Type 175 due to the use of metal adhesives?

The discussion on peeling seemed to show that there was no clear understanding of the physical nature of the bond which was made between the metals, and hence it was not possible to devise the critical test. Did the work at Bristol show any correlation of the peeling strength with the thickness of adhesive between the metallic surfaces? Mr. Parker had said that when the surfaces were pressed together in bonding by normal press, the adhesive was squeezed out and the peeling strength was very low. Was there a critical depth of adhesive and, if so, could this not be used as the check in place of the peeling test?

Mr. Parker had said that he had obtained perfectly sound joints using a pressure of only 20 to 25 lb./in.^{2}. Mr. Moss had thought it would be imprudent to think of anything less than 100 lb./in.^{2} pressure for production at this stage. It seemed that this was rather an over-cautious attitude and that if the lower pressure were unacceptable, then proof of this should be brought forward as early as possible. Clearly, any development that simplified the process of metallic bonding and made it cheaper was a real step forward at this stage and should be encouraged.

Mr. Parker: He had not emphasised reduction in cost due to the early days of production on his Company's plant, but he was confident that appreciable cost savings would be achieved on relatively long-run production because a bonding process required structures to be made up in sub-panel units. He thought that this would lead to useful cost savings, particularly on final assembly. Man power was definitely lower than for comparable riveted or bolted assembly. It was well known that on this type of work fatigue life was increased and there was the indirect cost saving due to reduction in weight.

It was estimated that on the Bristol Type 175 the major components into which Redux bonded items were built had a gross weight of 31,000 lb. The Redux bonded items themselves weighed 4,000 lb. and had Redux not been used the alternative design would have caused a weight increase of 1,000 lb.

He agreed with Mr. Rowe that there was no real knowledge of the physical nature of the bond. It had been fairly authoritatively stated that there was nothing complicated in it such as inter-molecular adhesion, but that metal bonding materials were simply good adhesives on the properly prepared material.

There were at least two useful papers on the subject of the physics of adhesion—"Strength of Glued Joints," *Aircraft Engineering*, Volume 16, 1944, by Dr. de Bruyne, and the Third Report from Aero

*Normally the Discussion following a Section Lecture is not published but on this occasion a written contribution enabled the Lecturer to give additional information in reply.

Research Ltd. to the Ministry of Supply on "The Peeling Strength of Bonded Light Alloy Strips," by C. Mylonas. His own Company had maintained the peel test, because it gave indication of process variations. On the other hand experience might prove that it was really a too critical selector. It was true to say that there was correlation between peel strength and the thickness of the adhesive, peel strength increasing with glue thickness. The opinion of his Company was that for optimum conditions of both shear and peel strength a glue line thickness of between 0.008 to 0.012 in. was the ideal. This dimension was often used on laboratory investigations, but some form of controlled breaking test such as the peel test was still required, as there was the possibility of weakness in a component due to lack of adhesion or cohesion (*e.g.* due to pre-treatment or materials) which would not affect the joint thickness.

He was quite satisfied that good bonds could be made with pressures as low as 2 to 25 lb./in.^{2} provided that the materials being bonded were either thin enough or were mated so well that the whole of the pressure exerted on the resin during the cure was not expended in forcing the parts into mating contact. This would rarely occur on structures of any size or strength; therefore he agreed with Mr. Moss that at the present stage of development the 100 lb./in.^{2} curing pressure should still apply as far as the press was concerned, but he thought that on the autoclave, by experience and in relation to the component being glued and by an examination of the glue line thickness pressures probably as low as 50 to 60 lb./in. could often be used.

A METHOD FOR DESIGNING WIND TUNNEL CONTRACTIONS

by

R. HARROP

SUMMARY

The contraction of a wind tunnel should be free from adverse pressure gradients, since this might cause boundary layer separation.

A wall contour has been designed for a circular cross-section contraction using incompressible flow theory. This gave a favourable pressure gradient at the beginning of the contraction where separation is likely to occur.

Appendix I compares the theory with experimental results obtained from a model of a proposed supersonic tunnel of which the contraction is rectangular in cross-section and which has been based on the results obtained in this report.

1. INTRODUCTION

Smith and Wang[1], using the analogy which exists between classical electricity and magnetism theory and hydrodynamical theory, have designed contraction cones.

If equal vortex rings replace the coils of a Helmholtz Galvanometer (*see* Section 3) a uniform distribution of velocity is obtained over a large part of the field in the centre of the coils, corresponding to the uniform magnetic field of the previous case. The lines of force of the magnetic case are now replaced by streamlines.

Thus it is necessary to find the value of the stream function of the flow consisting of the two vortex rings and a uniform flow parallel to the axis and then to find the equation of a streamline by making this function a constant. The surface swept out by this streamline on rotation round the axis is the boundary of the contraction cone.

The contraction ratio is determined by the strength of the vortices, while the length of the cone is fixed by the particular streamline chosen.

The velocity distribution along the wall is then calculated and the pressure gradient found.

NOTATION

The following are the conventions used in the application of vectors,

Scalar product	$\mathbf{A} \cdot \mathbf{B}$
Vector product	$\mathbf{A} \wedge \mathbf{B}$

All vectors are in Clarendon type. Other notation is as follows:

Vortex Ring

a	radius
Γ	circulation
$d\sigma$	filament cross-sectional area
$\boldsymbol{\zeta}$	vorticity strength
$\mathbf{B}_P$	vector potential at P due to ring
$d\mathbf{s}_o$	element of length of ring
θ	angular co-ordinate of point on ring

Paper received September 1950.

Mr. Harrop is at present on research at Caius College, Cambridge; the work described in the paper was done while he was in the Aerodynamics Department, Royal Aircraft Establishment.

General

$d\tau$	element of volume
$\mathbf{q}_P$	velocity at P
r_P	modulus of distance QP
$\mathbf{r}_1$	unit vector in direction **QP**
c_P	distance of P from axis
ψ_P	stream function at P
$d\mathbf{s}'$	element of arc of circle in plane perpendicular to axis, centre on axis and radius c_P
$d\mathbf{S}$	element of area of this circle
$2b$	separation of vortex rings
U_0	velocity at infinity on axis
z_P	axial co-ordinate of P

The following quantities were defined from those above:

$$C_P = c_P/a,\ Z_P = z_P/a,\ B = b/a$$

$$k^2 = \frac{4ac_P}{(a+c_P)^2+z_P^2} = \frac{4C_P}{(1+C_P)^2+Z_P^2}$$

for a vortex ring at $Z_P = 0$.

K, E are used to denote the complete elliptic integrals of the first and second kind respectively with arguments k in each case.

$$K = \int_0^{\frac{\pi}{2}} \frac{d\alpha}{(1-k^2\cos^2\alpha)^{\frac{1}{2}}},\quad E = \int_0^{\frac{\pi}{2}} (1-k^2\cos^2\alpha)\,d\alpha$$

Suffixes "1" and "2" were used for the two vortex rings.

In the Appendix

p	static pressure
H	total head
ρ	air density
V	air flow velocity

2. STREAM FUNCTION AND VELOCITY DISTRIBUTION FOR A VORTEX RING

The vorticity $\boldsymbol{\zeta}$ in a stream is defined by the relationship

$$\boldsymbol{\zeta} = \text{curl}\ \mathbf{q}$$

where $\mathbf{q}$ is the velocity at the point in the stream.

The vector potential $\mathbf{B}$ is defined by the equations

$$\left.\begin{aligned} \text{curl}\ \mathbf{B} &= \mathbf{q} \\ \text{div}\ \mathbf{B} &= 0 \\ \text{and} \qquad \mathbf{B} &= O(1/r) \text{ at infinity.} \end{aligned}\right\}$$

It will be noticed that this uses the fact that div $\mathbf{q} = 0$, the equation of continuity (since div curl $\mathbf{B} \equiv 0$).

This gives $\boldsymbol{\zeta} = \text{curl curl}\ \mathbf{B} = \text{grad div}\ \mathbf{B} - \nabla^2\mathbf{B}$

$$\nabla^2\mathbf{B} = -\boldsymbol{\zeta}.$$

The solution of this equation is

$$\mathbf{B}_P = \frac{1}{4\pi}\int \frac{\boldsymbol{\zeta}_Q\,d\tau}{r_P}$$

where the integral is taken over all points Q in the field of vorticity, $\mathbf{B}_P$ being the vector potential at P and r_P the modulus of the distance PQ.

This formula is given in Ref. 2.

Consider a point P and a vortex ring centre O, radius a and circulation Γ.

If $d\sigma$ is the area of cross-section of the filament and ζ the vorticity strength

$$\Gamma = |\boldsymbol{\zeta}|\, d\sigma$$

by definition.

Thus for the vortex ring

$$\mathbf{B}_P = \frac{\Gamma}{4\pi} \oint \frac{d\mathbf{s}_Q}{r_P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where $d\mathbf{s}_Q$ is the vector element of the ring and the integral is taken round the circumference, a result obtained in Ref. 2.

The velocity $\mathbf{q}_P$ is thus given by

$$\mathbf{q}_P = \operatorname{curl}_P \mathbf{B}_P = \frac{\Gamma}{4\pi} \oint \operatorname{curl}_P \left(\frac{d\mathbf{s}_Q}{r_P} \right)$$

$$= \frac{\Gamma}{4\pi} \oint \left[\operatorname{grad}_P \left(\frac{1}{r_P} \right) \wedge d\mathbf{s}_Q \right]$$

using the formula

$$\operatorname{curl} \phi \mathbf{A} = \phi \operatorname{curl} \mathbf{A} - \mathbf{A} \wedge \operatorname{grad} \phi$$

valid for any scalar ϕ and vector $\mathbf{A}$.

This equals $$\frac{\Gamma}{4\pi} \oint d\mathbf{s}_Q \wedge \operatorname{grad}_Q \left(\frac{1}{r_P} \right) = \frac{\Gamma}{4\pi} \oint d\mathbf{s} \wedge \frac{\mathbf{r}_1}{r_P^2} \quad . \quad . \quad . \quad (2)$$

where $\mathbf{r}_1$ is a unit vector in the direction from Q to P.

Consider the boundary circle radius c_P (the distance of P from the axis) and in a plane parallel to the vortex ring. Then if ψ_P is the stream function at P, the flux across the surface is

$$2\pi\psi_P = \int_{\substack{\text{area of} \\ \text{circle} \\ c_P}} \operatorname{curl} \mathbf{B}_P \,.\, d\mathbf{S}$$

$$= \oint_{\substack{\text{circle} \\ c_P}} \mathbf{B}_P \,.\, d\mathbf{s}' \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $d\mathbf{s}'$ represents an element of this circle, and ψ_P is thus given by

$$\psi_P = \frac{1}{2\pi} \oint \mathbf{B}_P \,.\, d\mathbf{s}' \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Take P as the point $(c_P, 0, z_P)$ in Cartesian co-ordinates and let θ be the angle shown in Fig. 1, then, using unit vectors $(\mathbf{i}, \mathbf{j}, \mathbf{k})$ along the axes,

$$d\mathbf{s}_Q = (a \cos\theta\, \mathbf{j} - a \sin\theta\, \mathbf{i})\, d\theta$$

and $$\mathbf{B}_P = \frac{\Gamma}{4\pi} \int_0^{2\pi} \frac{a\, d\theta\, (\cos\theta\, \mathbf{j} - \sin\theta\, \mathbf{i})}{r_P},$$

where $$r_P^2 = z_P^2 + a^2 + c_P^2 - 2ac_P \cos\theta\,.$$

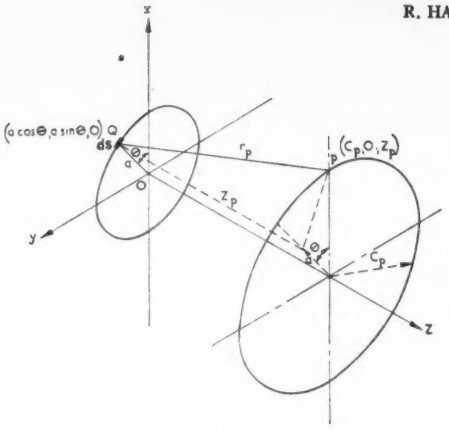


Fig. 1.
Flow through a vortex ring.

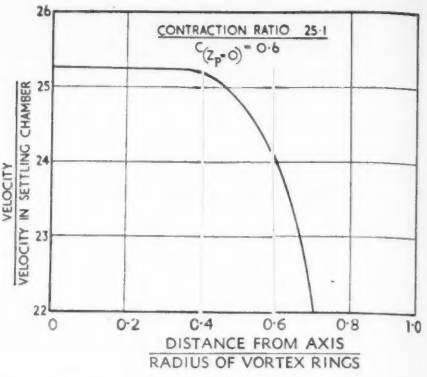


Fig. 2.
Velocity distribution across throat.

By symmetry the coefficient of **i** is zero

and
$$\mathbf{B}_P = \frac{\Gamma}{4\pi}\int_0^{2\pi} \frac{a\cos\theta}{r_P}\,\mathbf{j}\,d\theta,$$

giving
$$\psi_P = \frac{\Gamma a c_P}{4\pi}\int_0^{2\pi} \frac{\cos\theta}{r_P}\,d\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

By transformation
$$r_P^2 = \left[(a+c_P)^2 + z_P^2\right]\left[1 - k^2\cos^2\frac{\theta}{2}\right]$$

where
$$k^2 = 4ac_P/[(a+c_P)^2 + z_P^2]$$

and
$$\psi_P = \frac{\Gamma a c_P}{4\pi\left((a+c_P)^2+z_P^2\right)^{\frac{1}{2}}}\int_0^{2\pi} \frac{\cos\theta\,d\theta}{\left(1-k^2\cos^2\frac{\theta}{2}\right)^{\frac{1}{2}}}$$

or
$$\psi_P = \frac{a\Gamma}{\pi k}\left(\frac{c_P}{a}\right)^{\frac{1}{2}}\left[\left(1-\frac{k^2}{2}\right)K - E\right]$$

where $K = \int_0^{\frac{\pi}{2}} \frac{d\alpha}{(1-k^2\cos^2\alpha)^{\frac{1}{2}}}$ and $E = \int_0^{\frac{\pi}{2}} (1-k^2\cos^2\alpha)^{\frac{1}{2}}\,d\alpha$

that is K and E are complete elliptic integrals of the first and second kinds respectively. (Ref. 3, page 85.)

These formulae are obtained by an alternative method in Ref. 4.

It is convenient at this point to relate c_P and z_P to the radius a of the vortex ring thus

$$C_P = c_P/a,\ Z_P = z_P/a$$

$$\psi = \frac{\Gamma a (C_P)^{\frac{1}{2}}}{\pi k}\left[\left(1-\frac{k^2}{2}\right)K - E\right] \quad . \quad . \quad . \quad . \quad (6)$$

where
$$k^2 = \frac{4C_P}{[(1+C_P)^2 + Z_P^2]}.$$

From equation (2)

$$\text{Velocity} = \frac{\Gamma}{4\pi}\oint ds \wedge \frac{\mathbf{r}_1}{r_P^2}.$$

This is the Biot-Savart formula and is an alternative starting point for the vortex theory.
The velocity thus equals

$$\frac{\Gamma}{4\pi}\int_0^{2\pi}\frac{(-\sin\theta,\ \cos\theta,\ 0)\wedge(c_P - a\cos\theta,\ -a\sin\theta,\ z_P)\,a\,d\theta}{r_P^3} \quad . \quad . \quad . \quad (7)$$

The z component of this is

$$\frac{\Gamma}{4\pi}\int_0^{2\pi}\frac{(a-c_P\cos\theta)}{r_P^3}a\,d\theta$$

$$= \frac{\Gamma}{4\pi}\int_0^{2\pi}\frac{(a-c_P\cos\theta)\,a\,d\theta}{\left[(a+c_P)^2+z_P^2\right]^{\frac{3}{2}}\left[1-k^2\cos^2\frac{\theta}{2}\right]^{\frac{3}{2}}}.$$

which can be expressed as

$$\frac{\Gamma}{4\pi}\left\{\frac{a^2}{\left[(a+c_P)^2+z_P^2\right]^{\frac{3}{2}}} - \frac{ac_P}{\left[(a+c_P^2)+z_P^2\right]^{\frac{3}{2}}}\left(\frac{2}{k^2}-1\right)\right\}\int_{\theta=0}^{\theta=2\pi}\frac{d\theta}{\left[1-k^2\cos^2\frac{\theta}{2}\right]^{\frac{3}{2}}} +$$

$$+\frac{\Gamma}{4\pi}\left\{\frac{ac_P}{\left[(a+c_P)^2+z_P^2\right]^{\frac{3}{2}}}\right\}\frac{2}{k^2}\int_{\theta=0}^{\theta=2\pi}\frac{d\theta}{\left[1-k^2\cos^2\frac{\theta}{2}\right]^{\frac{1}{2}}}.$$

Using the formula

$$\int_0^{\frac{\pi}{2}}\frac{d\alpha}{(1-k^2\cos^2\alpha)^{\frac{3}{2}}} = \frac{E}{1-k^2}$$

on page 56 of Ref. 3, and substituting for $[(a+c_P)^2+z_P^2]$,

$$\frac{\Gamma}{4\pi}\left\{\frac{k^3a^{\frac{1}{2}}}{2c_P^{\frac{3}{2}}} - \frac{k^3}{2a^{\frac{1}{2}}c_P^{\frac{1}{2}}}\left(\frac{2}{k^2}-1\right)\right\}\frac{E}{1-k^2} + \frac{\Gamma}{4\pi}\frac{k}{a^{\frac{1}{2}}c_P^{\frac{1}{2}}}K.$$

This can be written as

$$\frac{\Gamma}{\pi a}\left\{\frac{k^3}{8C_P^{\frac{3}{2}}} - \frac{(2k-k^3)}{8C_P^{\frac{1}{2}}}\right\}\frac{E}{1-k^2} + \frac{\Gamma}{\pi a}\frac{k}{4C_P^{\frac{1}{2}}}K \quad . \quad . \quad . \quad (8)$$

From the equation (7) it is seen that the x component of velocity is given by

$$\frac{\Gamma}{4\pi}\int_0^{2\pi}\frac{az_P\cos\theta\,d\theta}{r_P^3}$$

$$= -\frac{\Gamma}{4\pi}\frac{z_Pa}{\left[(a+c_P)^2+z_P^2\right]^{\frac{3}{2}}}\left\{\frac{2}{k^2}\int_{\theta=0}^{\theta=2\pi}\frac{d\theta}{\left[1-k^2\cos^2\frac{\theta}{2}\right]^{\frac{1}{2}}} - \left(\frac{2}{k^2}-1\right)\int_{\theta=0}^{\theta=2\pi}\frac{d\theta}{\left[1-k^2\cos^2\frac{\theta}{2}\right]^{\frac{3}{2}}}\right\}.$$

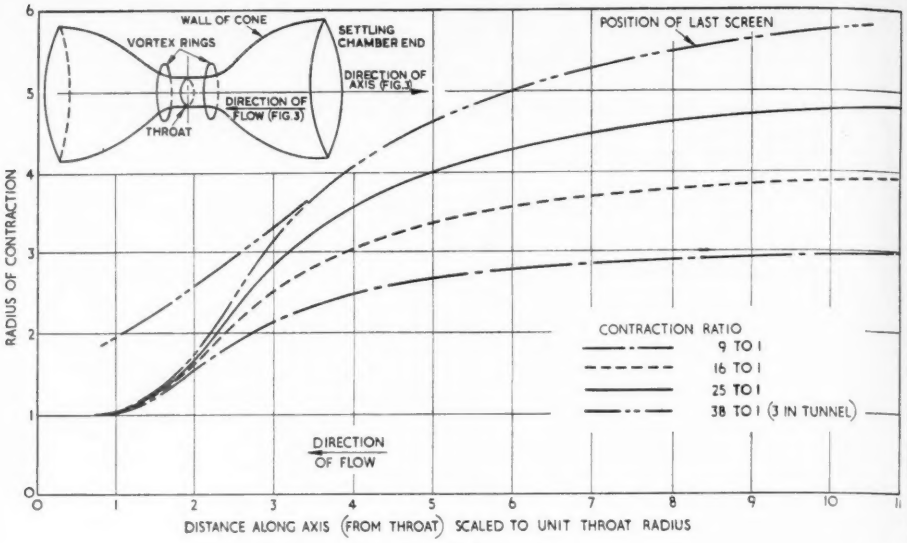


Fig. 3.
Contraction cones using $C_{(Z_P=0)}=0.6$.

Substituting for the integrals and $[(a+c_P)^2+z_P{}^2]^{\frac{3}{2}}$ the x component of the velocity can be expressed as

$$\frac{\Gamma}{8\pi}\frac{z_P k^3}{a^{\frac{1}{2}}c_P{}^{\frac{3}{2}}}\left\{-\frac{2K}{k^2}+\left(\frac{2}{k^2}-1\right)\frac{E}{1-k^2}\right\}$$

or

$$\frac{\Gamma}{8\pi a}\frac{Z_P k^3}{C_P{}^{\frac{3}{2}}}\left\{-\frac{2K}{k^2}+\left(\frac{2}{k^2}-1\right)\frac{E}{1-k^2}\right\} \quad . \quad . \quad . \quad (9)$$

3. DESIGN OF CONTRACTION

In the Helmholtz Galvanometer[5] there are two equal coils placed at $z=\pm b$ while there is a uniform magnetic field (that of the earth) parallel to the axis. On changing to the hydrodynamical analogy it is necessary to include a uniform stream corresponding to this field in order that the streamlines may eventually become parallel to the axis.

The expressions for ψ and the velocity due to the rings are obtained by substituting (z_P+b) for z_P for the first ring which is denoted by a suffix "1" and (z_P-b) for z_P for the second ring which is denoted by a suffix "2," in the results just obtained.

If U_0 is the velocity at infinity and $b/a=B$, then from equation (6) the value of the stream function ψ is

$$\psi=\frac{a\Gamma}{\pi k_1}(C_P)^{\frac{1}{2}}\left[\left(1-\frac{k_1{}^2}{2}\right)K_1-E_1\right]+\frac{a\Gamma}{\pi k_2}(C_P)^{\frac{1}{2}}\left[\left(1-\frac{k_2{}^2}{2}\right)K_2-E_2\right]+\frac{1}{2}U_0 C_P{}^2 a^2$$

where

$$k_1{}^2=\frac{4C_P}{(1+C_P)^2+(Z_P+B)^2}$$

and a similar expression exists for $k_2{}^2$.

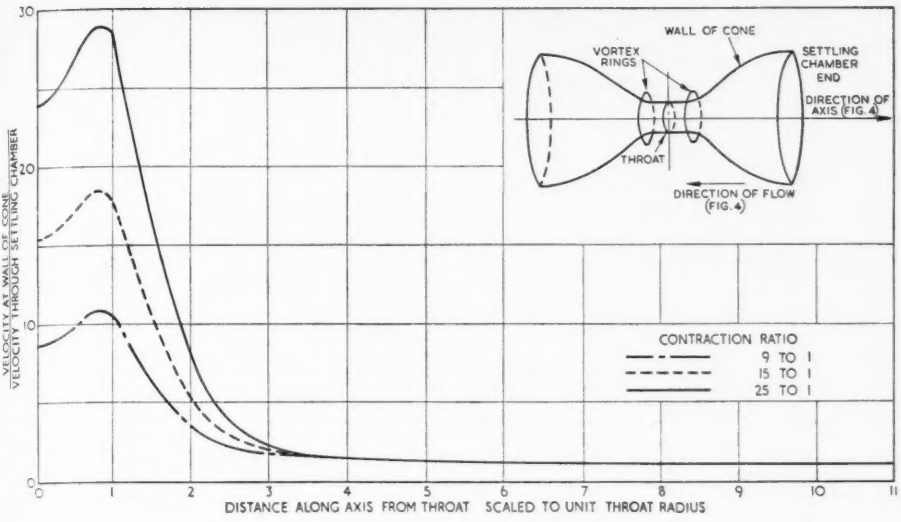

WALL OF CONE
VORTEX RINGS
SETTLING CHAMBER END
DIRECTION OF AXIS (FIG. 4)
THROAT
DIRECTION OF FLOW (FIG. 4)
CONTRACTION RATIO
9 TO 1
15 TO 1
25 TO 1
VELOCITY AT WALL OF CONE / VELOCITY THROUGH SETTLING CHAMBER
DISTANCE ALONG AXIS FROM THROAT SCALED TO UNIT THROAT RADIUS

Fig. 4.
Velocity distribution in contraction cone.

Therefore

$$\frac{\psi}{U_0 a^2} = \frac{\Gamma}{U_0 \pi a}\frac{(C_P)^{\frac{1}{2}}}{k_1}\left[\left(1-\frac{k_1^2}{2}\right)K_1 - E_1\right] + \frac{\Gamma}{U_0 \pi a}\frac{(C_P)^{\frac{1}{2}}}{k_2}\left[\left(1-\frac{k_2^2}{2}\right)K_2 - E_2\right] + \frac{1}{2}C_P^2$$

As in Ref. 1 the value of B is taken as 0.46936.

Given the contraction ratio and the value of C_P for the point on the streamline where $Z_P=0$, the value of C_P is known for the point where $Z_P=\infty$ and since the vortex rings have no effect there

$$\Gamma/(U_0 a^2)=\tfrac{1}{2}C^2_{(Z_P=\infty)}$$

and the value of ψ is known for the whole streamline.

Putting $A=\Gamma/(U_0 a\pi)$

$$\frac{\psi}{U_0 a^2} = A\frac{(C_P)^{\frac{1}{2}}}{k_1}\left[\left(1-\frac{k_1^2}{2}\right)K_1 - E_1\right] + A\frac{(C_P)^{\frac{1}{2}}}{k_2}\left[\left(1-\frac{k_2^2}{2}\right)K_2 - E_2\right] + \frac{1}{2}C_P^2 .$$

The conditions at $Z_P=0$ determine A.

The co-ordinates of points on the streamline are found by fixing Z_P and calculating $\psi/(U_0 a^2)$ for several estimated values of C_P. Interpolation then enables the correct value to be found since it corresponds to $\psi/(U_0 a^2)=\frac{1}{2}C^2_{(Z_P=\infty)}$

The choice of C_P for $Z_P=0$ was made as a result of considerations of the length of the contraction and of the uniformity of the resulting velocity distribution. The greater the value of C_P at this point the shorter the contraction but the worse the distribution of the velocity at the throat.

This is illustrated in Fig. 2 which shows the velocity distribution expressed by the ratio (velocity at the point in question)/(velocity in settling chamber) for a contraction ratio of 25:1 and with $C_P=0.6$ initially.

(The drawing is continued outside the $C_P=0.6$ line in order to show the effect of using a higher value for this constant. It will be observed that the effect of

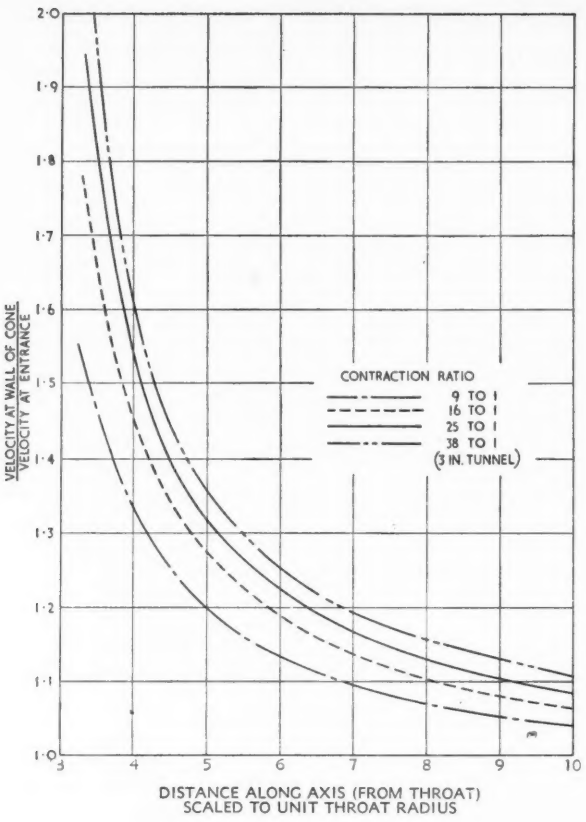


Fig. 5.
Velocity distribution near entrance

choosing a contraction ratio of 25:1 and an initial C_P of 0.6 has been to fix the strength of the vortex rings so that the mean velocity across the part from $C_P=0$ to $C_P=0.6$ is 25 times that of the stream at infinity.)

In consequence a value of $C(z_{P=\infty})$ of 0.6 was used and wall profiles calculated for contraction ratios of 9:1, 16:1 and 25:1.

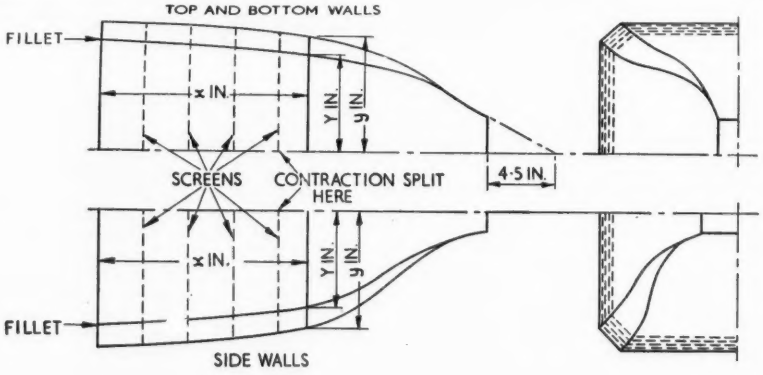
The results, scaled to unit throat radius, are plotted in Fig. 3.

The additional curve shown in that figure relates to the model tunnel and is explained in the Appendix to the report.

4. CALCULATION OF THE VELOCITY DISTRIBUTION

From equations (8) and (9) is obtained the velocity given by an axial component of

$$U_0\left\{1+A\left[\left\{\frac{k_1^3}{8C_P^{\frac{3}{2}}}-\frac{2k_1-k_1^3}{8C_P^{\frac{1}{2}}}\right\}\frac{E_1}{1-k_1^2}+\frac{k_1K_1}{4C_P^{\frac{1}{2}}}+\right.\right.$$
$$\left.\left.+\left\{\frac{k_2^3}{8C_P^{\frac{3}{2}}}-\frac{2k_2-k_2^3}{8C_P^{\frac{1}{2}}}\right\}\frac{E_2}{1-k_2^2}+\frac{k_2K_2}{4C_P^{\frac{1}{2}}}\right]\right\}$$



TOP AND BOTTOM WALLS

x IN.	WALL y IN.	FILLET Y IN.
0	9·25	7·75
2·0	9·14	7·64
4·0	9·01	7·51
6·0	8·91	7·41
8·0	8·75	7·25
10·0	8·56	7·06
12·0	8·30	6·81
14·0	7·99	6·58
16·0	7·58	6·28
18·0	7·01	5·91
20·0	6·21	5·43
22·0	5·04	4·74
24·0	3·61	3·61
26·0	2·42	—

SIDE WALLS

x IN.	WALL y IN.	FILLET Y IN.
0	9·25	7·75
2·0	9·14	7·64
4·0	9·01	7·51
6·0	8·91	7·41
8·0	8·75	7·25
10·0	8·56	7·06
12·0	8·30	6·81
14·0	7·92	6·51
16·0	7·24	5·94
18·0	6·12	5·02
20·0	4·36	3·62
22·0	2·83	2·55
24·0	1·84	1·84
26·0	1·50	—

Fig. 6.
Co-ordinates of model contraction.

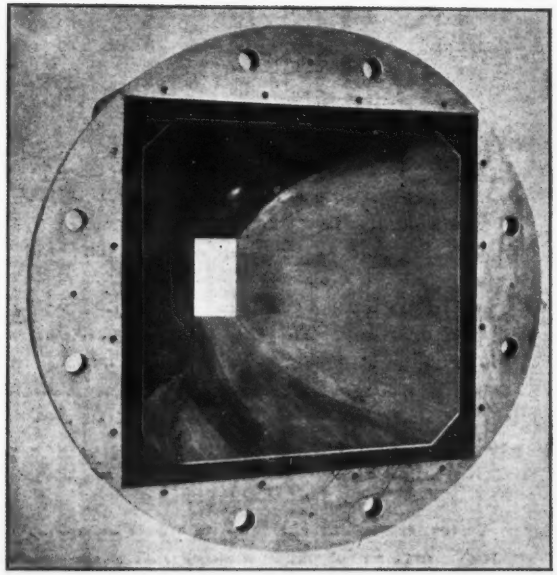


Fig. 7.
Model contraction.

and a radial component of

$$U_0 A \left\{ \frac{(Z_P + B)}{8C_P^{\frac{5}{2}}} \left[\frac{k_1 (2 - k_1^2)}{1 - k_1^2} E_1 - 2K_1 k_1 \right] + \frac{(Z_P - B)}{8C_P^{\frac{3}{2}}} \left[\frac{k_2 (2 - k_2^2)}{1 - k_2^2} E_2 - 2K_2 k_2 \right] \right\} .$$

From these two expressions the velocity distribution has been computed and plotted in Figs. 4 and 5, the contraction being scaled to have unit throat radius.

It is seen that at the entrance to the contraction the pressure gradient is everywhere favourable.

An alternative method of design for axi-symmetrical contractions is given in Ref. 6

APPENDIX I

TESTS MADE ON MODEL CONTRACTION

The contraction is a 1/12 scale model of that of the 3 ft. supersonic tunnel to be erected at the National Aircraft Establishment, Bedford.

It has two pairs of symmetrical walls, the top and bottom walls finishing at an angle of about 28 degrees to the centre line, the side walls becoming parallel before the end.

Screens are fixed at 3 in. intervals for the first 12 in. as shown in Fig. 6. At distances of 2 in. starting 2 in. after the last of these screens, pressure holes have been placed in the centres of the top and side walls and also in two of the fillets. These holes can be seen in Fig. 7 which is a photograph of the contraction.

Figure 8 shows the working section of the tunnel. The top liner is removable and several different ones have been designed for a range of Mach numbers. Each

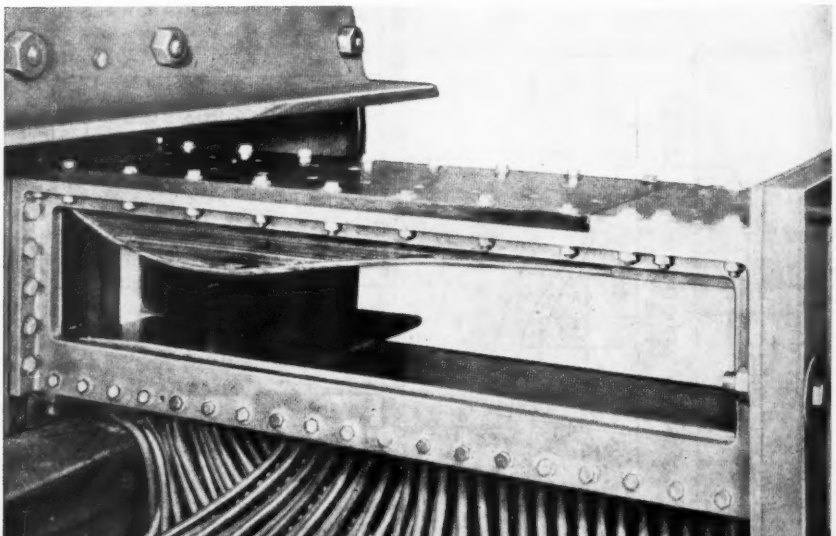


Fig. 8.
Working section of 3 in. tunnel.

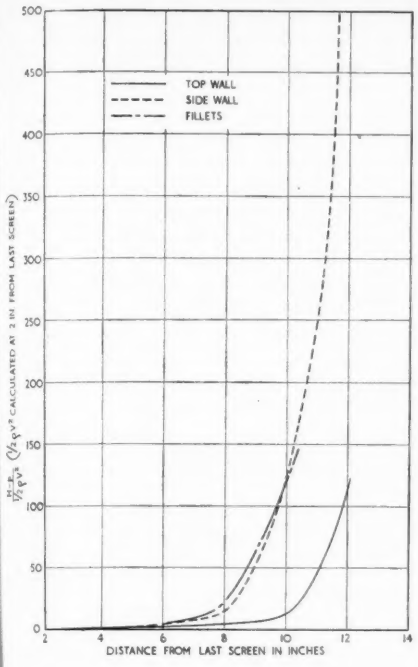


Fig. 9.
Pressure distribution in model contraction.

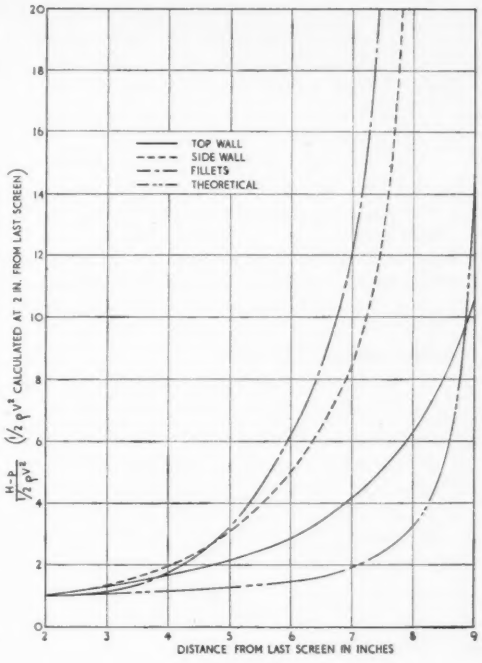


Fig. 10.
Pressure distribution in model contraction.

of these, and the fixed bottom wall fair in with the wall of the fixed contraction. A report will be issued on the design of, and the experimental work done on the liners.

In Fig. 3 the mean wall of the contraction (based on area considerations) is plotted, together with an estimated position for the theoretical curve. An estimated theoretical velocity distribution for the latter curve is shown in Fig. 5.

It was not known what the effect of changing from a circular to a rectangular contraction would be but it was expected that adverse pressure gradients might arise on the fillets. Experimental work was done therefore in which the total head was measured together with static pressure at the pressure holes. For this purpose the tunnel was not attached and a pitot tube was securely fastened at the exit to the contraction.

Figures 9 and 10 show the results obtained together with the theoretical distribution previously mentioned. Considerable difficulty was found in measuring some of the pressure differences which were very small. It was found that the velocity distribution at the entrance was not uniform, the initial velocity being smaller on the side wall than on the top wall and smaller still on the fillet. In consequence it was decided to use for the term $\frac{1}{2}\rho V^2$ the value corresponding to the first pressure hole on each respective wall. Further, since the pressure distributions on the two fillets should be the same, the mean of the experimental results was used.

The following conclusions may be drawn from the results of the tests:—

(*i*) In spite of the change to rectangular cross-section the contraction still has a positive pressure gradient at all points.

(*ii*) The experimental values of $(H - p)/(\frac{1}{2}\rho V^2)$ are higher than the theoretical ones at the start of the contraction. This is probably due to the change from the theoretical profile which occurs at about 7 or 8 inches from the last screen.

(*iii*) The velocity increase along the side wall is greater than along the top wall, this probably being caused by the fact that the inclinations and curvatures on the first wall are greater than on the second wall.

The direct computation of the shape of a rectangular (or even square) contraction is considerably more difficult than that of a circular one, and it was for this reason that this theoretical method was computed in spite of the fact that in practice a rectangular shape is generally required. The experimental work done on the model contraction seems to indicate that the change of shape, at least in this case, did not cause the occurrence of any places of adverse pressure gradient.

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A NOTE ON THE BUCKLING OF STRUTS

by

HAROLD LURIE, Ph.D.

SUMMARY

Invariably the texts in elasticity and strength of materials derive the stability criteria for struts by first setting up the beam bending moment equation. Together with suitable boundary conditions this equation is then solved as an eigenvalue problem for non-zero solutions. It is indicated here that in general this procedure is too limited, both physically and mathematically, and that in some cases the correct approach yields additional results which normally would not become evident by employing the classical method.

INTRODUCTION

A brief review of the nature of the problem will serve as a preliminary basis for further discussion. If a straight bar is subjected to an axial thrust, it can undergo only axial compression. This is true for any value of the thrust, provided that the bar is "straight" and the thrust "axial" in the exact sense. However, the stability of the bar may be affected: for all values of the thrust up to a "critical" value, the bar will return to its original straight form after it has been given a slight displacement. But for certain values of the thrust the bar becomes unstable and will not return to its original form. This can be shown by energy considerations.

Looking at the problem from a slightly different point of view, the values of the axial thrust which are consistent with equilibrium in the slightly displaced form can be determined. This leads to the so-called eigenvalue problem. A homogeneous differential equation containing a parameter (in this case the axial thrust) together with homogeneous boundary conditions, yields non-zero solutions only for discrete values of the parameter. In this problem these values will furnish the critical loads.

The usual method of setting up the differential equation is to use the moment equation

$$EI\,d^2y/dx^2 = -M(x). \quad . \quad (1)$$

Depending upon the end fixity, the bending moment M is determined as a function of x. This equation is then solved, and the boundary conditions provide the equation for the determination of the non-zero values of the parameter.

Paper received October 1950.

Dr. Lurie was formerly Lecturer in Aeronautics, California Institute of Technology, Pasadena, California, and is now with The Rand Corporation, Santa Monica, California.

NOTATION

E	modulus of elasticity
I	moment of inertia
x	co-ordinate measured along the strut
y	lateral deflection of strut
M	bending moment
P	axial thrust
P'	axial thrust in Fig. 4(*b*)
w	transverse load per unit length
k	$\sqrt{[P/(EI)]}$
l	length of strut
n	integer
k_n	value of k corresponding to n $(=2n\pi/l)$
ϕ_n	$k_n l/2$
A,B,C,D	constants of integration
P_1	Euler critical load $(=\pi^2EI/l^2)$
P_{cr}	critical load
M_0	bending moments at ends of strut in Fig. 4(*a*), in opposite directions.
M_0'	bending moments at ends of strut in Fig. 4(*b*), in the same direction
R	shear reactions at ends of strut in Fig. 4(*b*)

BASIC METHOD OF SOLUTION

For any beam problem, four boundary conditions are always available—two at each end. For a column subjected to end thrust only, the boundary conditions are:

(*a*) pinned end: $y=0;\ d^2y/dx^2=0$. (2)

(*b*) fixed end: $y=0;\ dy/dx=0$. (3)

(*c*) free end: $\dfrac{d^2y}{dx^2}=0;\ EI\dfrac{d^3y}{dx^3}+P\dfrac{dy}{dx}=0.$ . . . . (4)

Being a second order differential equation, equation (1) is too limited to provide a general solution. Instead, the fourth order loading equation

$$\frac{d^2}{dx^2}\left(EI\frac{d^2y}{dx^2}\right)=w(x) \quad . \quad (5)$$

should be used. This makes the problem mathematically consistent, and also allows a general equation to be set up which is independent of the boundary conditions. This is important, because it allows an equation to be formulated which is independent of the mode of buckling. The only assumption regarding the mode is that equilibrium is possible for some shape other than the original straight form, provided the displacements are small. For large displacements this differential equation is no longer valid and the exact curvature expression would have to be employed.

To use equation (5), the axial force P must be replaced by an equivalent transverse load $w(x)$. The axial force induces a bending moment Py in the deflected column, and the load corresponding to this bending moment distribution $M(x)$ is equal to

$$w(x)=-d^2M/dx^2.$$

Hence the equivalent lateral load which would produce a moment Py is equal to

$$w(x)=-P\,d^2y/dx^2.$$

Therefore equation (5) becomes

$$\frac{d^2}{dx^2}\left(EI\frac{d^2y}{dx^2}\right)=-P\frac{d^2y}{dx^2}$$

or, for a uniform beam,

$$\frac{d^4y}{dx^4}+k^2\frac{d^2y}{dx^2}=0 \quad . \quad . \quad (6)$$

where $k^2=P/EI.$

Equation (6), together with the relevant boundary conditions (2), (3) or (4), should be employed for the general solution of the column problem.

EXAMPLE

A good example of the principles involved is provided by the case of a column fixed at both ends. Without considering any specific mode of buckling, the problem can be stated mathematically as the differential equation (6) together with the boundary conditions (3), *i.e.*

$$(y)_{x=0}=(y)_{x=l}=0;\ \left(\frac{dy}{dx}\right)_{x=0}=\left(\frac{dy}{dx}\right)_{x=}=0.$$

From equation (6),

$$d^2y/dx^2=A\cos kx+B\sin kx.$$

Therefore

$$y=-(A/k^2)\cos kx-(B/k^2)\sin kx+Cx+D.$$

Putting in the boundary conditions at $x=0$

$$0=-A/k^2+D$$

$$0=-B/k+C.$$

Therefore

$$y=(A/k^2)(1-\cos kx)+$$
$$+(B/k^2)(kx-\sin kx). \quad (7)$$

Putting in the boundary conditions at $x=l$,

$$0=(A/k^2)(1-\cos kl)+(B/k^2)(kl-\sin kl)$$
. . . . (8)

$$0=(A/k)\sin kl+(B/k)(1-\cos kl).$$
. . . . (9)

For a solution, the determinant must be equal to zero.

Therefore

$$(1-\cos kl)^2-(kl-\sin kl)\sin kl=0$$

i.e. $$2-2\cos kl-kl\sin kl=0. \quad . \quad (10)$$

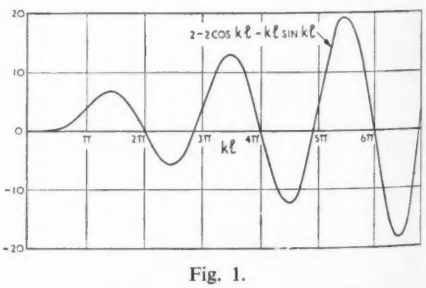

Fig. 1.
Graphical representation of the complete characteristic equation.

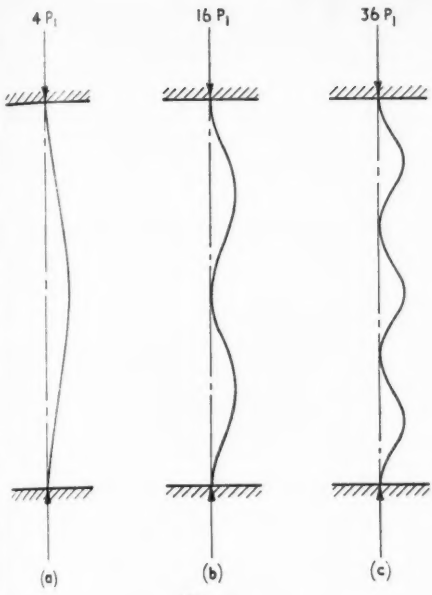

Fig. 2.
Buckling modes corresponding to the solution of $\sin(kl/2)=0$.

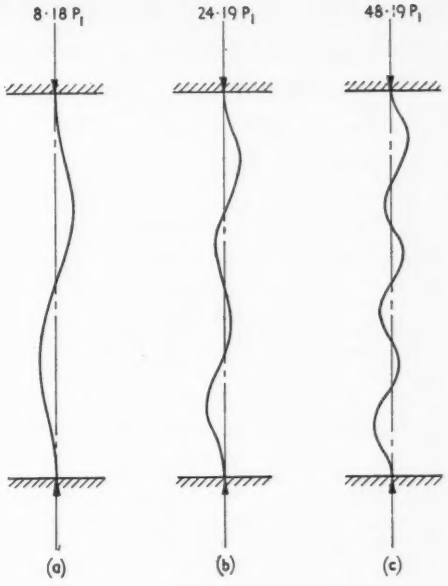

Fig. 3.
Buckling modes corresponding to the solution of $\tan(kl/2)=kl/2$.

The solution of this transcendental equation will give the values of k leading to the critical loads. Fig. 1 shows a plot of this equation, the zeros corresponding to the eigenvalues. Note that there is a sequence of solutions alternating with the classical solution of $kl=2\pi n$.

The two sequences can be separated by rewriting equation (10) as

$$\sin(kl/2)\,[2\sin(kl/2)-kl\cos(kl/2)]=0$$

Therefore either

$$\sin(kl/2)=0 \quad . \quad . \quad (11)$$

or

$$\tan(kl/2)=kl/2. \quad . \quad (12)$$

Equation (11) yields the usual values of the critical load, namely

$$P_{cr}=4\pi^2n^2EI/l^2. \quad (n=1,2,3,\ldots) \quad (13)$$

Equation (12) yields an additional sequence the first three values of which are

$$kl/2=4.494,\ 7.725,\ 10.904,\ldots \quad (14)$$

Therefore

$$P_{cr}=\frac{8.18\pi^2EI}{l^2},\ \frac{24.19\pi^2EI}{l^2},\ \frac{48.19\pi^2EI}{l^2},\ \ldots$$
$$. \quad . \quad . \quad . \quad (15)$$

The shape of the modes can be obtained directly by the usual methods. First consider the eigenfunctions obtained from equation (11). This equation leads to the eigenvalues

$$kl/2=n\pi.$$

Therefore $k_n=2n\pi/l.$

Substituting these values of k in equation (8)

$$0=(A/k_n^2)(1-\cos 2n\pi)+$$
$$+(B/k_n^2)(2n\pi-\sin 2n\pi)$$
$$=(B/k_n^2)(2n\pi)$$

Hence $B=0.$

Therefore equation (7) becomes

$$y=(A/k^2)[1-\cos(2n\pi x/l)]. \quad . \quad (16)$$

These modes are shown in Fig. 2, in which P_1 represents the Euler critical load π^2EI/l^2. They are the modes corresponding to the classical critical loads. Note that as the square bracket in equation (16) is always positive, the deflections must necessarily lie on one side of the undeflected shape.

Next consider the eigenfunctions obtained from equation (12). Let the sequence of numerical values for $kl/2$ be ϕ_n (*See* equation (14)).

Therefore $k_n = 2\phi_n / l$.

Substitute these values of k in equation (9)

$$0 = (A/k_n)\sin 2\phi_n + (B/k_n)(1 - \cos 2\phi_n)$$

Therefore $A/B = -\tan \phi_n$.

Equation (7) becomes

$$y = C\left[\frac{2\phi_n x}{l} - \sin\frac{2\phi_n x}{l} - \tan\phi_n\left(1 - \cos\frac{2\phi_n x}{l}\right)\right].$$

These modes are shown in Fig. 3, in which again P_1 represents the Euler critical load. They are similar to the modes of a strut of which both ends are constrained to remain on a fixed line, one end being "fixed" and the other "pinned."

An important practical point immediately becomes apparent. Suppose such a clamped strut is to be used to carry a load greater than the lowest buckling load, and this is to be accomplished by restraining the centre from lateral displacement. According to the classical formula, the strut will then carry a load of $16\pi^2 EI/l^2$ ($n=2$ in equation (13)). Actually it will carry only $8.18\pi^2 EI/l^2$ and buckle into the shape shown in Fig. 3(*a*). This can be shown to represent a lower energy state than the mode of Fig. 2(*b*). Similarly, for all modes higher than the fundamental, the beam may buckle in the series given by Fig. 3 rather than Fig. 2 if only the stationary points are restrained from lateral displacement. Hence, from a practical point of view the classical solution is important for the fundamental mode only. Any additional lateral restraint could make the column buckle according to Fig. 3.

DISCUSSION OF RESULTS

It is instructive now to reconsider the whole problem, and to determine in just what measure the classical approach fails to yield the complete solution.

From a mathematical point of view, the reason has already been stated. A fourth order differential equation must always be used in order to satisfy the four possible boundary conditions of a beam. From a physical point of view, the limitation was imposed on the original diagram from which

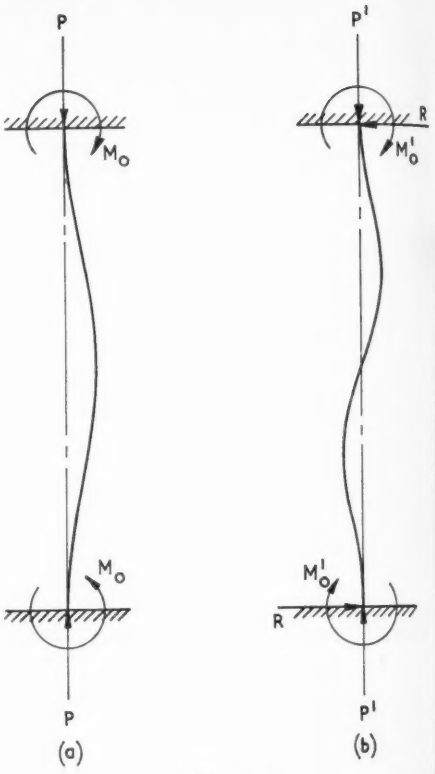

Fig. 4.
External forces and moments for the two types of buckling.

the forces and moments were obtained. Fig. 4(*a*) shows this complete diagram. Certainly the classical solution gives the correct result for this mode, and further, for all similar modes.

But consider a mode of the second sequence, as shown in Fig. 4(*b*). Here the end moments are in the same direction, and shear reactions are necessary for equilibrium. If these reactions are included in the second order moment equation, this second type of mode will become apparent—but not the first! Only the fourth order equation will yield all possible modes.

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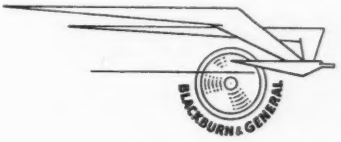

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